THE AMERICAN SOCIETY OF
MECHANICAL ENGINEERS

TRANSACTIONS

VOLUME 43

CHICAGO MEETING
NEW YORK MEETING
1921

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THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

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JOHN L. HARRINGTON, *Constitution and By-Laws*
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[1] Elected 1921 to fill unexpired term of J. R. Allen, deceased.

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[1] Partial list of Society Representatives.

SUMMARY OF MEMBERSHIP BY RESIDENCE

United States

Alabama	96	Nebraska	25
Alaska	2	Nevada	4
Arizona	22	New Hampshire	48
Arkansas	16	New Jersey	955
California	503	New Mexico	8
Canal Zone	4	New York	3706
Colorado	77	North Carolina	51
Connecticut	719	North Dakota	4
Delaware	103	Ohio	1096
District of Columbia	191	Oklahoma	83
Florida	40	Oregon	36
Georgia	103	Pennsylvania	1631
Hawaiian Islands	17	Philippine Islands	18
Idaho	5	Porto Rico	7
Illinois	968	Rhode Island	153
Indiana	259	South Carolina	25
Iowa	41	South Dakota	5
Kansas	51	Tennessee	61
Kentucky	42	Texas	122
Louisiana	83	Utah	17
Maine	37	Vermont	37
Maryland	185	Virginia	102
Massachusetts	1310	Washington	86
Michigan	534	West Virginia	42
Minnesota	132	Wisconsin	312
Mississippi	11	Wyoming	6
Missouri	238		
Montana	18	Total	14487

Foreign Countries

Africa	16	Italy	7
Australia	18	Jamaica	1
Belgium	5	Japan	25
Canada	218	Mexico	30
Central America	4	Norway	7
China	29	Roumania	2
Cuba	57	Russia	5
Czechoslovakia	1	Santo Domingo	4
Denmark	4	Singapore Straits Settlements	2
Dutch East Indies	3	South America	65
Finland	4	Spain	5
France	41	Sweden	10
Germany	19	Switzerland	7
Great Britain	104	Trinidad	2
Greece	1	Turkey	3
Holland	2		
India	17	Total in Foreign Countries	718

Summary of Membership by Residence

Membership in United States	14487
Membership in Foreign Countries	718
Present Address Unknown	22
Total Membership	15227

Membership by Grades

Honorary Members	19
Members	7332
Associates	753
Associate-Members	3505
Juniors	3618
Total	15227

TRANSACTIONS

OF

The American Society of Mechanical Engineers

VOLUME 43 — 1921

VOLUME 43 contains the papers and addresses presented at the Spring and Annual Meetings of The American Society of Mechanical Engineers for the year 1921, the discussion thereon and important addresses. In selecting the material published in the present volume, the intention has been to include all of the papers possessing permanent reference value. A number dealing more particularly with industrial and economic matters and whose interest and value depended to a great extent upon the timeliness of their presentation have been omitted, but are listed in the index.

EDWIN S. CARMAN

Edwin S. Carman was born in Prairie Depot, Ohio, in 1878. His high-school and business education was supplemented by special instruction and studies in engineering at the Central Manual Training School at Cleveland, Ohio.

He began his work in the shop of the Sun Oil Company, of Toledo, Ohio, and four years later entered the engineering field with the American Machine and Manufacturing Company, of Cleveland, Ohio. He was appointed chief engineer after two years in the engineering department, and while with the company he designed a complete line of electric traveling and special forge-shop cranes and hoists, billet rotators, wire-fence machinery, rolling-mill equipment, etc. In 1908 this company was consolidated with the Johnston and Jennings Company, of Cleveland, Ohio, and Mr. Carman was appointed chief engineer and manager of the engineering and machine department.

The Osborn Manufacturing Company, of Cleveland, Ohio, an

old-established and well-known house, manufacturers of and dealers in wire, bristle and fiber brushes and wheels, hardware specialties, foundry supplies and equipment, early recognized the possibilities of making foundry molds by machine power and decided to enter this promising field. In 1908, Mr. Carman was engaged by this company to design, manufacture and build up a complete line of foundry molding machines, the details of manufacture in regard to workmanship equal to that of the machine-tool industry.

In 1913 Mr. Carman became directly associated with the Osborn Manufacturing Company as chief engineer in charge of engineering and manufacture of the machine division, and in 1916 he was elected secretary and appointed works manager of both the machine and brush divisions of the company.

He is the author of a treatise on Foundry Molding Machines and Pattern Equipment and a contributor of papers on the art of machine molding.

For a number of years Mr. Carman has been prominent in the activities of engineering societies. After serving it in several capacities, he was elected President of the Cleveland Engineering Society, and completed his service in this office in June 1920. He was first chairman of the Cleveland Section of The American Society of Mechanical Engineers, which was authorized in December 1918. Mr. Carman was a member of the A.S.M.E. Committee on Aims and Organization and Chairman of Sub-Committee C, which dealt with relations of the mechanical engineers to other engineers. He was one of the Society's representatives on the Joint Conference Committee. This Committee is representative of the A.S.C.E., A.I.M.E., A.S.M.E., and A.I.E.E. and as a result of the activity he displayed in the work of this Committee he was selected as its spokesman to address the joint meeting of members of the governing boards of the several societies which met at Chicago in April 1920. He was subsequently appointed one of the A.S.M.E. delegates to the Washington Conference on June 3–4, which resulted in the organization of The Federated American Engineering Societies. In December 1919, he was appointed chairman of the A.S.M.E. Committee on Local Sections, with a seat on the Council. Mr. Carman has taken a very active part in the affairs of the Society since his election to membership in 1917, and has also found time and opportunity to contribute to the advancement of his local and state engineering organizations as well as to participate in the national federation development.

No. 1784

SOCIETY AFFAIRS

THE SPRING MEETING

Chicago, Ill., May 23 to 26

The attendance at the Spring Meeting, held at the Congress Hotel, Chicago, May 23 to 26, exceeded that of any previous Spring Meeting, the total registration being 1235 members and guests. The program consisted of fourteen sessions with numerous committee meetings and plant visits. The first morning was devoted to registration and to the Council Meeting, the Business Session being held in the afternoon, and on the following days the mornings were devoted to professional sessions and the afternoons to plant visits. Tuesday evening was devoted to the Session on Training in the Industries.

An informal reception was held in the Gold Room of the Congress Hotel on Monday evening, at which the Chairman of the Chicago Section and the President of the Western Society of Engineers, with the officers of the Society, greeted members at the convention. An informal dance was held Wednesday evening. Special attention was accorded the ladies attending the convention in the form of tours to the Marshall Field store and to the Field Museum, while on Wednesday, following an automobile ride through the residential section of Chicago, they were guests at the home of Mrs. R. W. Hunt.

The plants visited by members of the Convention included the Fisk Street Station of the Commonwealth Edison Company, Sears, Roebuck & Company, Illinois Steel Company, Pennsylvania Terminal, Crane Company, Mandel Brothers, Chicago Mill and Lumber Company, Western Electric Company, Pullman Company, Chicago Underwriters Laboratory and the Yellow Cab Manufacturing Company.

Technical features of the meeting included the Fuel Session on Tuesday morning at which six papers were presented, and the Management Session at which was presented the report of the work of

the Executive Committe during the first year of the organization of the management division. Robert Thurston Kent presented a progress report of the joint committee on Management Terminology. These reports were followed by the presentation of three papers on the subject of management.

At the first General Session were presented papers on power plant subjects which were quite extensively discussed.

For the first time in the history of the Society a session was devoted to the subject of training for the industries.

There was a session devoted to the consideration of the engineering problems of Chicago involved in the transportation and terminal questions. F. K. Copeland, President of the Western Society of Engineers, presided at this meeting.

At the second General Session a number of papers on general subjects were presented. The papers by R. S. Johnston and W. M. White were productive of considerable discussion. The Power Test Codes Committee held a public hearing for the purpose of discussing the proposed A.S.M.E. Power Test Codes on general construction, reciprocating steam engines and evaporating apparatus. The first two codes were adopted with such slight verbal changes as the Committee might find were necessary for the sake of consistency. The third code was referred back to the individual committee having it in charge for consideration at the conference to be held later with a similar committee of the American Institute of Chemical Engineers.

Four widely discussed papers were presented at the Railroad Session; and at the Materials Handling Session, the subject for its first meeting was the importance of machinery and its intelligent use in building roads. The Forest Products Session was arranged by the organizing committee of the newly formed Forest Products Division.

At the Power Session two papers on the subject of power in the Middle West were presented.

Prior to the convention the Society visited McCook Field on May 21. This was in accordance with a plan instituted by the new Aeronautics Division of the Society to develop the interest of the membership and the engineering profession in general in the national aviation policy both governmental and commercial. The Society of Automotive Engineers also participated in this visit. The Air Service had prepared an elaborate program for the entertainment and instruction of the members of the Society.

Subsequent to the convention about forty members went to Rock Island joining with some sixty engineers from the Tri-Cities Section for the events at Rock Island and Davenport. The events commenced with an inspection of the Rock Island arsenal which was followed by a luncheon, after which the opening session of the Ordnance Division was held. A program of five technical papers was presented. In the evening the members attended a dinner served at the Hotel Blackhawk.

The entire arrangement of the Chicago Meeting was under the jurisdiction of the General Chicago Committee made up of Robert W. Hunt, Honorary Chairman, Herbert S. Philbrick, Chairman, James D. Cunningham, Secretary, Melville S. Flinn, Treasurer, David Lofts, P. Albert Poppenhusen, Arthur L. Rice and Edward P. Rich.

PROGRAM

Monday Morning, May 23

Council Meeting.
Meeting of Main Committee on Power Test Codes.

Monday Afternoon

Meeting of Power Test Codes Committee on Steam Turbines.
Business Meeting.
ON THE ORGANIZATION OF AN ENGINEERING SOCIETY, Morris L. Cooke.[1]
Proposed New Constitution of the Society.
Reports of Committees on Technical Nomenclature, on Steel Roller Chains, and on the Standard Tonnage Basis for Refrigeration.

Monday Evening

Informal Reception.

Tuesday Morning, May 24

Meeting of Local Sections Committee.
Meeting of Committee on Weights and Measures.
Local Sections Conference.

[1] Published in MECHANICAL ENGINEERING, May 1921.

SIMULTANEOUS SESSIONS

FUEL

Recording Ash-Pit Loss From Chain-Grate Stokers, E. G. Bailey.

Boiler Tests with Pulverized Coal, Henry Kreisinger and John Blizard.

Limitations of Mechanical Stokers Utilizing Mid-West Coals, E. H. Tenney.

Capacity and Efficiency Limitations of Stokers Using Mid-west Coals, John E. Wilson.

A Discussion of the Subject of Smoke and Its Relation to the General Health of the Community, Dr. John Dill Robertson.

Latest Requirements of the City of Chicago in Furnace Design with Special Reference to Hand-Fired Boilers and Limits of Each Design, Frank Chambers.

MACHINE SHOP PRACTICE

Influence of the Automobile on the Machine Tool Industry in General, F. K. Hendrickson.[1]

Influence of the Automobile on Gear Cutting and Gear Cutting Machinery, Henry J. Eberhardt.[1]

The Relation of Power Presses and Dies to the Automobile Industry, Henry J. Hinde.[1]

Influence of the Automobile on Lathe Practice, R. E. Flanders.[1]

Fundamentals of Interchangeable Manufacture, C. B. Lord.

MANAGEMENT

Progress Report of Management Division Executive Committee.

Report of Committee on Management Terminology.

Graphical Methods, W. C. Marshall.

Industrial Waste, L. W. Wallace.

GENERAL

Capacity Tests of Dry-Vacuum Pumps by the Low-Pressure Nozzle, Snowden B. Redfield.

Report Upon Efficiency Tests of a 30,000-kw. Steam Turbine, H. B. Reynolds.

Tuesday Afternoon

Session on Training for the Industries.

What the National Metal Trades Association Is Doing and Intends to Do in Industrial Education, H. C. Smith.[1]

The Engineering Industries and Education, D. C. Jackson and M. W. Alexander.[2]

General Education and the Engineering Profession, H. E. Miles[1]

[1] Published in Mechanical Engineering, August 1921.

[2] Published in Mechanical Engineering, June 1921.

Wednesday Morning, May 25

SIMULTANEOUS SESSIONS

CHICAGO

Some Aspects of the Problem of Chicago as Mid-Interior Rail-Water Gateway, J. R. Bibbins.

Factors in Terminal Capacity and Development:

Development of Air Rights in Connection with City Freight Houses, E. J. Noonan.

Mechanical Plan for Modern Types of Local Freight Houses for Large Centers, J. D'Esposito, E. H. Lee and Harwood Frost.

Freight Movement by Motor Trucks — Viewpoint of Carrier and Public, Hugh E. Young.

Freight Tunnel System as a Terminal Distribution Agency, J. R. Bibbins and E. J. Noonan.

Terminal Unification, J. H. Brinkerhoff.

The Function of the Terminal Survey, J. R. Bibbins.

Relation of Steam Roads to Rapid Transit Development, Bion J. Arnold.

GENERAL

Investigation of Oxy-Acetylene Welding and Cutting Blowpipes, R. S. Johnston.

Interpretation of Boiler-Water Analyses, J. R. McDermet.

The Hydraucone Regainer, Its Development and Applications in Hydroelectric Plants, W. M. White.

POWER TEST CODES

Discussion of Proposed Power Test Codes on General Instructions, Reciprocating Steam Engines and Evaporating Apparatus.

Wednesday Afternoon

Conference of Student Branches.

Meeting of Standing Committee on Professional Divisions.

Wednesday Evening

Informal Dance.

Thursday Morning, May 26

SIMULTANEOUS SESSIONS

RAILROAD

Advantages of Large Freight Locomotives, Particularly the 2-10-2 Type, Albert F. Stuebing.

The Design of Large Locomotives, M. H. Haig.

Necessity for Improvements in Design and Operation of Present-Day Locomotives, H. W. Snyder.

MATERIALS HANDLING

PLANNING AND ORGANIZING A ROAD JOB FOR MECHANICAL HANDLING OF MATERIALS, C. D. Curtis.[1]

ROAD-CONSTRUCTION PLANTS, B. H. Piepmeier.[1]

MECHANICAL NEEDS IN HIGHWAY-CONSTRUCTION MACHINERY, R. C. Marshall, Jr.[1]

FOREST PRODUCTS

THE PAPER INDUSTRY OF AMERICA DEPENDENT UPON A PERMANENT WOOD SUPPLY, Hugh P. Baker.

POWER

LOCATION AND DISTRIBUTION OF CENTRAL STATION POWER IN THE MIDDLE WEST, W. L. Abbott.

FUTURE POWER DEVELOPMENT IN THE MIDDLE WEST, C. W. Place.

Thursday Afternoon

MOTION PICTURES

MODERN CONCRETE ROAD CONSTRUCTION, with explanatory remarks, by E. H. Lichtenberg.

METHODS AND MACHINERY USED IN CONSTRUCTING AND MAINTAINING EARTH, GRAVEL, MACADAM, AND RIGID-SURFACE ROADS, with explanatory remarks, by William Ord.

THE ANNUAL MEETING

The forty-second Annual Meeting of the Society was held in New York, December 5 to 9, 1921. Notwithstanding the general situation in the engineering industries, the registration was 1854, comparing favorably with that of previous years.

The program was arranged so that professional sessions were held on Monday and Tuesday and on Thursday and Friday with an all-day business meeting on Wednesday.

Of the nineteen professional sessions nine were conducted by the new Professional Divisions.

The Annual Conference of Local Sections Delegates was attended by representatives of forty-four local sections, and besides considering organization problems of the local sections, it devoted a considerable amount of time to the proposed constitution of the Society. At the business meeting the new constitution and two amendments

[1] Published in MECHANICAL ENGINEERING, June 1921.

to the present Constitution were considered. It was voted to refer the amendments relating to the voting of Junior members and the mechanism of amending the Constitution to the membership by letter ballot, and the proposed contsitution was referred to the Constitution and By-Laws Committee for further revision. The first award of the A.S.M.E. medal was made to Hjalmar G. Carlson for his invention and part in the production of 20,000,000 Mark III drawn-steel booster casings used principally as a component of 75-mm. high-explosive shells, but also extensively in gas shells and bombs. The annual prizes for best papers were also presented at this session. The Junior Prize was awarded to S. Logan Kerr, for his paper on the Moody Ejector Turbine, and Student Prizes were awarded to Karl H. White for his paper on Forces in Rotary Motors, and to R. H. Morris and A. J. R. Houston for their Report on Investigation of Herschel Type of Improved Weir.

Twenty-seven committee meetings were scheduled on the program and at least a dozen others were held impromptu.

The leading social event was the Dinner-Dance at the Hotel Astor on Thursday evening. The Smoker for the men this year was held at the Fifth Avenue Restaurant. The President's Reception and the Ladies Tea were held on the fifth floor of the Engineering Societies Building.

The formal excursions included visits to the Seaboard By-Product Coke Company, the Essex Street Station of the Public Service Corp. of N. J., The Davis-Bournonville Company, the Ford Motor Company, the oil-burning plant of the Singer Building and the S.S. *Olympic* of the White Star Line.

On Tuesday evening after an inspiring address in which he portrayed the Society as having completed its first cycle and emerging into a new era, President E. S. Carman relinquished the chair in favor of Dean Dexter S. Kimball, whom the membership had selected to head the Society during 1922.

A further ceremony of the evening was the bestowing of Honorary Membership in the Society on Past-President Henry R. Towne. Following the exercises in the Auditorium the company adjourned to the reception rooms where the new officers received the members of the Society and the ladies. Then came the customary informal get-together, with the renewal of old associations and the formation of many new ones. Music was provided, and the evening ended with an informal dance.

Under the auspices of the Committee on Relations with Col-

leges and under the direct supervision of Dr. H. G. Tyler of this committee, the members of the Student Branches of the Society conducted a session on Thursday morning. An innovation in the meeting activities of the Society and in its relations with students this session considered two technical papers and a statement of the problems of Student Branch operation in a manner equal to if not better than some of the sessions conducted by mature members of the Society.

A better interchange of scientific and research information through the agency of the Engineering Division of the National Research Council is within reach if the ideas developed at the Research Conference of the Annual Meeting and transmitted to Mr. A. D. Flinn, vice-chairman of the Division, who was present, are carried out.

This conference, held Thursday, December 8, was conducted by Prof. A. M. Greene, Jr., chairman of the Society's Research Committee, and the attendance of over forty included many authorities on engineering research. Two sessions were held, culminating in the organization of an Advisory Committee to help the National Research Council in the correlation of research data as well as to make suggestions to hasten the completion of the Council's scheme.

The Society for the Promotion of Engineering Education joined Thursday afternoon with the A.S.M.E. in a session of three papers on the relations between engineering education and the industries.

There were many addresses of an instructive and inspiring nature which it has been impossible to include in Transactions. At the opening session on Elimination of Waste in Industry, L. P. Breckenridge and W. S. Murray spoke of the work of the Superpower Survey, and E. F. DuBrul directed attention to the tremendous wastes in industry caused by industrial depression in business cycles. At the Machine Shop session J. J. Callahan gave an inspiring address. Past-President Charles T. Main gave a report on the second World Cotton Conference held in England last summer which he attended as a delegate of the Textile Division of the Society. The Ordnance Division, which held its first session at this meeting, was opened by an address by Gen. Guy E. Tripp, Chairman of the Board of the Westinghouse Electric and Manufacturing Company, who emphasized the responsibility of the Ordnance Division. Col. J. W. Joyes, Chief of the Technical Staff, Ordnance Department, U. S. A., presented a paper on the record and policies of the Army Ordnance Department in regard to the elimination of waste.

On Thursday afternoon R. Sanford Riley presented moving pictures showing actual conditions in a stoker-fired furnace. The great interest aroused in this exhibit rendered its repetition necessary.

The chairmen of the Committees having in charge the arrangements for the Annual Meeting were: Roy V. Wright, Chairman of Committee on Meetings and Program, L. B. McMillan, General Chairman of Sub-Committee for 1921 Annual Meeting, Gustave R. Tuska, Chairman, President's Reception, F. F. Uehling, Chairman, Acquaintanceship, C. P. Bliss, Chairman, Courtesy, W. S. Bowen, Chairman, Information, J. I. Lyle, Chairman, Informal Get-Together and Smoker, Edw. Van Winkle, Chairman, Dinner Dance, H. D. Edwards, Chairman, Excursions, R. G. Macy, Chairman, Catering, Mrs. Frank T. Chapman, Chairman, Ladies' Tea, Mrs. Nixon Lee, Chairman, Ladies' Acquaintanceship and Miss Burtie Haar, Chairman, Ladies' Excursions.

PROGRAM

Monday Morning, December 5

Council Meeting
Conference of Local Sections' Delegates
Boiler Code: Public Hearings on Air Tanks and Pressure Vessels

Monday Afternoon

Session on Elimination of Waste in Industry, addresses by Sir Herbert Llewellyn Smith, K. C. B., Fred J. Miller and L. P. Breckenridge.

Monday Evening

SESSION ON EDUCATION AND TRAINING IN THE INDUSTRIES

Outline of Education and Training in the Industries, R. L. Sackett.[1]
Education and Training on Railroads, D. C. Buell.[1]

Tuesday Morning, December 6

SIMULTANEOUS SESSIONS

POWER WASTE

Auxiliary System and Heat Balance at the Delaware Station of the Philadelphia Electric Company, E. L. Hopping.

Heat Balance of Colfax Station, C. W. E. Clarke.

Heat Balance System for Hell Gate Station, J. H. Lawrence and W. M. Keenan.

Heat Balance of the Connors Creek Plant of the Detroit Edison Company, C. Harold Berry and F. E. Moreton.

[1] Published in Mechanical Engineering, February 1922.

MACHINE SHOP WASTE

Waste in Machine Industry, J. J. Callahan.
Salvaging Industrial Wastes, J. A. Smith.[1]
On the Art of Milling, John Airey and Carl J. Oxford.

RAILWAY

Avoidable Waste in Operation of Locomotives and Cars, Wm. Elmer.
Avoidable Waste in Locomotives, Jas. Partington.
Avoidable Waste in Car Operation — The Container Car, Walter C. Sanders.

Tuesday Afternoon

SIMULTANEOUS SESSIONS

GAS POWER

Porting and Charging of Two-Stroke Oil Engines, Louis Illmer.
Compounding the Combustion Engine, E. A. Sperry.

FOREST PRODUCTS WASTE

Address by David L. Goodwillie, Chicago, Ill., Chairman, U. S. Chamber of Commerce Committee on Conservation and Reforestization,[2] and by J. H. Wallace on Paper and By-Products from Southern Pine Refuse.[2]

MANAGEMENT WASTE

Making Work Fascinating, W. N. Polakov.[3]
Process Charts, F. B. and L. M. Gilbreth.
The Rochester Shoe Wage Arbitration, Sanford E. Thompson.
Reports of Committees on Mangement Terminology and Measuring Managerial Ability.

Tuesday Evening

Presidential Address, Edwin S. Carman.
Conferring Honorary Membership upon Henry R. Towne, Chairman of Board, Yale and Towne Manufacturing Company.
Reception.

Wednesday Morning, December 7

BUSINESS MEETING

Reports of Standing Committees; Reports of Special Joint Committee with S. A. E. on Steel Roller Chains; Discussion of Amendments to Constitution; Second Reading of Proposed New Constitution, Award of Prizes.

[1] Published in Mechanical Engineering, December 1921.
[2] Published in Mechanical Engineering, March 1922.
[3] Published in Mechanical Engineering, November 1921.

Wednesday Afternoon

BUSINESS MEETING

Discussion of Proposed New Constitution.
Ladies Tea and Reception.

Wednesday Evening

Smoker.

Thursday Morning, December 8

SIMULTANEOUS SESSIONS

FUEL WASTE

Boiler Plant Efficiency, Victor J. Azbe.
Boiler and Furnace Design, D. S. Jacobus.
Fuel Saving in Relation to Capital Necessary, Jos. Harrington.
Fuel Saving in Modern Gas Producers and Industrial Furnaces, W. B. Chapman.

MATERIALS HANDLING WASTE

Material Handling an Important Factor in the Elimination of Industrial Waste, H. V. Coes.[1]

STUDENT SESSION

Discussion of Draft-Tube Design with Reference to the Hydraucone, W. K. Ramsey.
Flow of Water in Hydraulic Turbine Tubes, Geo. E. Lyon.
The Armour Idea of the Student A.S.M.E.

RESEARCH CONFERENCE

Elimination of Waste in Industry Through Research, F. A. Wardenburg.[2]
Research in Leather Manufacture, Arthur W. Thomas.[2]
Report of Research Sub-Committee on Lubrication.
Discussion of Steam Table Research.

Thursday Afternoon

SIMULTANEOUS SESSIONS

PROFESSIONAL ENGINEERING EDUCATION FOR THE INDUSTRIES

[1] Published in Mechanical Engineering, December 1921.
[2] Published in Mechanical Engineering, February 1922.

Joint Session with S.P.E.E.

PROFESSIONAL ENGINEERING EDUCATION FOR THE INDUSTRIES, F. C. Pratt[1]
A NATIONAL POLICY ON ENGINEERING EDUCATION, A. G. Christie.[1]
COLLEGE EDUCATION AS RELATED TO INDUSTRY, J. E. Otterson.[1]
THE ENGINEERING SCHOOL AND THE INDUSTRIES, Dexter S. Kimball.[1]

GENERAL

STRESSES AND DEFORMATION IN FLAT CIRCULAR CYLINDER HEADS, A MATHEMATICAL ANALYSIS, G. D. Fish.

MOTION PICTURES

CLOSE-UP OF STOKER COMBUSTION, R. Sanford Riley.[2]
HANDLING MATERIALS IN COAL AND LIMESTONE INDUSTRIES.

Thursday Evening

Dinner Dance.

Friday Morning, December 9

SIMULTANEOUS SESSIONS

AERONAUTIC WASTE

COMMERCIAL OPERATIONS OF AIRPLANES, L. B. Lent.
AIR LINES AND SOME OF THEIR PROBLEMS, R. B. C. Noorduyn.[2]
STUDY OF THE ELASTIC PROPERTIES OF SMALL-SIZE WIRE CABLE, R. R. Moore.
TESTS OF PLYWOOD WEBS WITH LIGHTENING HOLES ARRANGED AS IN AIRPLANE RIBS, D. T. Brown and R. J. Diefenbach.

TEXTILE WASTE

REPORT ON SECOND WORLD COTTON CONFERENCE, Chas. T. Main.
HIDDEN WASTES IN TEXTILE PLANTS, T. P. Gates.
ECONOMY IN TEXTILE DRYING, B. R. Andrews.

GENERAL

TESTING OF EMERGENCY FLEET BOILERS USING OIL FUEL, F. W. Dean.
STEAM CONDENSING PLANTS, Paul A. Bancel.
VERTICAL TRIPLE EXPANSION PUMPING ENGINE, L. A. Quayle and E. H. Brown.

ORDNANCE WASTE

THE RECORD AND POLICIES OF THE ARMY ORDNANCE DEPARTMENT IN REGARD TO ELIMINATION OF WASTE, Col. J. W. Joyes.
Address by General G. E. Tripp.

Friday Afternoon

Council Meeting.

[1] Published in MECHANICAL ENGINEERING, January 1922.
[2] Published in MECHANICAL ENGINEERING, February 1922.

REPORT OF THE VISIT TO ENGLAND AND FRANCE BY DELEGATES OF THE A.S.C.E., A.I.M.E., A.S.M.E. AND THE A.I.E.E.

The purpose of this visit was the personal expression of that intense feeling of admiration, sympathy and brotherhood which the engineers of America entertain generally for the engineers of Great Britain and France, admiration for their splendid and effective services in the winning of the Great Victory, sympathy for their losses on the battlefield, and a hearty desire to coöperate in any possible way in the solution of the great problems of the future and the presentation of the John Fritz Medal to Sir Robert Hadfield in London and to Mr. Charles Eugene Schneider in Paris.

The delegation consisted of twelve members, as follows:

Ambrose Swasey, *Chairman* Charles T. Main, *Chairman* John R. Freeman, *Past Vice-President*	Am. Soc. C. E.
Ira N. Hollis, *Chairman, Past President* Ambrose Swasey, *Past President* Jesse Merrick Smith, *Past President*	Am. Soc. M. E.
Colonel A. S. Dwight, *Chairman* Charles F. Rand, *Past President* William Kelly, *Member*	A. I. M. & M. E.
Dr. F. B. Jewett, *Chairman, Vice-President* Dr. A. E. Kennelly, *Past President* Major General George O. Squier, *Member*	A. I. E. E.

By reason of the fact that some of the representatives had membership in more than one Society, the A. S. M. E. had in the deputation five of its Past-Presidents as follows:

Ambrose Swasey
Jesse Merrick Smith
John R. Freeman
Charles T. Main
Ira N. Hollis

Colonel A. S. Dwight served as Vice-Chairman and Mr. Charles F. Rand as Honorary Secretary of the deputation.

The delegation was received most hospitably in both London and Paris, and a list of the several reunions is here given as a matter of record.

Monday, June 27: Dinner by Mr. Swasey to the delegates at the Hotel Cecil, in London.

Monday, June 27, 3:00 p.m.: Invitation from the Anglo-American Society to the inaugural lecture by the Right Honorable Viscount Bryce.

Tuesday, June 28: Visit to National Physical Laboratory in Teddington, on the invitation of the Royal Society. This laboratory corresponds to our Bureau of Standards in Washington. Dinner with the Council of the Institution

of Civil Engineers at Great George Street at 7:00 p.m. with addresses by the American Engineers and short responses by British Engineers. At 9:00 p.m. the dinner was adjourned for the James Forrest Lecture, Fuel Problems of the Future, delivered by Sir George T. Beilby, F.R.S.

Wednesday, June 29, 10:00 a.m.: Meeting of the Institution of Civil Engineers of Great Britain at the home of the institution on Great George Street: opening address by president of the institution. In the presence of a considerable assemblage of engineers and public men, greetings and good wishes from the American engineers were conveyed by Dr. Ira N. Hollis to the engineering profession of Great Britain. An illuminated address signed by the Presidents of the four Founder Societies was presented to the Civil Engineers as custodians for the profession in Great Britain. Mr. Ambrose Swasey on the part of the board of Award presented the John Fritz Medal to Sir Robert Hadfield for his eminent service to engineering. The American Ambassador, Mr. Harvey, and Viscount Bryce were present and the latter made an address.

Wednesday, June 29, 1:30 p.m.: Luncheon at Goldsmith's Hall with the Iron and Steel Institute, the Institution of Mining Engineers and the Institution of Mining and Metallurgy.

Wednesday, June 29, 8:00 p.m.: Dinner with the Smeatonian Society of Civil Engineers, the oldest known engineering society, founded in 1777, and now maintained as a sort of honorary society. Many distinguished men were present, including Sir James Dewar, inventor of liquid air, and Sir James Crichton Brown as guests of honor.

Thursday, June 30, 10:00 a.m.: Meeting with the Institution of Mechanical Engineers for general papers at their rooms on St. George Street. Mr. Swasey was elected to Honorary Membership in the British Institution of Mechanical Engineers.

Thursday, June 30, p.m.: Meeting with Council of the Royal Society at Burlington House. Informal reception followed by discussion of several papers.

Visit to House of Parliament, and tea with members of House of Commons on Thames Embankment.

Thursday, June 30, 7:00 p.m.: Dinner with the Council of the Institution of Electrical Engineers at the Royal Palace Hotel, Kensington, followed at 8:30 by a Conversazione and Reception by the President and Council at the Natural History Museum, South Kensington.

*Friday, July 1, a.m.:*Meeting of a small committee of the deputation with the Council of the Institution of Mechanical Engineers for the purpose of explaining to them the plan and functions of the United Engineering Society, our joint Library, The Engineering Foundation, and The Federated American Engineering Societies.

Friday, July 1, 6:45 p.m.: Dinner with the Council of the Institution of Mechanical Engineers at St. Stephen's Club.

Friday, July 1, 9:30 p.m.: Reception by Sir Robert and Lady Hadfield at 22 Carlton House Terrace.

There were other meetings in London of an unofficial character, the last of which was the usual dinner of The American Society in London on the 4th of July at the Hotel Cecil, at which the American ambassador and a number of other well-known persons spoke. This concluded the official stay in England and the members left in time for the Paris meeting.

Friday, July 8, noon, Paris: The members of the delegation and ladies accompanying them were invited to the Eiffel Tower for lunch at noon. They were met by M. Gustave Eiffel, who conducted them to his private suite at the top of the Tower, where light refreshments were offered accompanied by some graceful toasts, and afterwards brought them to the third story where they met at luncheon a number of representatives of the government and the Société des Ingénieurs Civils de France, places being set for about 100.

Friday, July 8, 8:30 p.m.: Meeting of the Civil Engineers of France at their society building. Greetings of the American engineers to the French engineers were conveyed by Dr. Ira N. Hollis in an impressive speech leading up to the delivery of an illuminated address signed by the presidents of the four American societies represented. A statement in French was made by Mr. Jesse Merrick Smith, Vice-President of The American Section of the Société des Ingénieurs Civils de France, of the meaning of the John Fritz Medal, followed by the presentation by Mr. Swasey of the John Fritz Medal to M. Charles Eugene Schneider. After a fitting response by M. Schneider, Mr. Swasey and Herbert Hoover were elected Honorary Members of the Civil Engineers of France and Mr. Swasey was decorated with the cross of an Officer of the Legion of Honor for his services to engineering, by a representative of the French Government.

The members of the delegation were invited by M. Schneider to visit the great steel plant at Creusot as his guests, and those who could accept gathered there from various side trips on July 22, and remained until the 24th, during which time they were entertained most delightfully at the Company's guest house. Formal luncheons presided over by executive officers of the company took place every day of the visit, and every opportunity was offered to inspect the extensive plant and auxiliary activities.

Two members of the delegation, Mr. Cummings and Mr. Freeman, were able to accept the invitation of the Société des Ingénieurs Civils de France to go with members of the Society on its annual excursion to inspect recent developments in the southeastern part of France. Development in metallurgy and hydroelectric power in the French Alps and the new harbor works at Marseilles were points of special interest on the trip.

The following honorary memberships were conferred on the members of the delegation in England and France:

Swasey:	Institution of Mechanical Engineers Institution of Mining Engineers Société des Ingénieurs Civils de France Legion of Honor — Officer
Rand:	The Iron & Steel Institute
Dwight:	Institution of Mining and Metallurgy
Kelly:	Institution of Mining Engineers
Edwin Ludlow:	President, American Institute of Mining and Metallurgical Engineer's was made an Honorary Member of the Institution of Mining and Metallurgy.
Herbert Hoover:	Société des Ingénieurs Civils de France.

No. 1785

RECORDING ASH-PIT LOSS FROM CHAIN-GRATE STOKERS

By E. G. Bailey, Cleveland, Ohio
Member of the Society

This paper presents certain preliminary data obtained from a device recently developed to record the ash-pit loss from chain-grate stokers and thus enable firemen to effectively control this factor.

The device consists of a steel tube or bulb filled with nitrogen and connected through a capillary tube to a recorder consisting of a mercury U-tube, one leg of which is open to the atmosphere and carries a float to which the recorder pen is attached. It has been found that when this bulb is properly located near the rear of a chain-grate stoker, its temperature will respond definitely to changes in the amount of combustiable going to the ash pit and will vary in direct proportion to the heat thus lost.

Records of ash-pit loss, excess air, unburned gas, flue-gas temperature, and boiler rating make it possible to plot characteristic performance curves for stokers showing the losses due to unburned combustible, unburned gases and excess air, and to determine the proper amount of excess air to be maintained and also the most efficient capacity. These curves can be used to compare the relative efficiencies of different types of stokers, the suitability of different kinds of coal, and the effectiveness of the control by the fireman.

THERE has been more guesswork and less knowledge regarding ash-pit loss than any other thing of equal importance in the boiler plant. It pays to watch the ash pile from every kind of stoker or even hand-fired furnaces, though it be only visual inspection to note whether it is well burned out or full of combustible. To sample and analyze the ashes regularly and calculate the heat lost in that manner is better still, yet it is done in comparatively few plants.

2 Even the sampling and analysis of the ashes fall far short of accomplishing the desired result. A sample taken from the entire plant each day leaves the responsibility divided between the different firemen and different types of stokers. To take a sample from each boiler for each shift would still not accomplish as good results as if the ash-pit loss were continuously recorded for each boiler.

Presented at the Spring Meeting, Chicago, Ill., May 23 to 26, 1921, of The American Society of Mechanical Engineers.

3 Instantaneous readings of any kind are most valuable in aiding the fireman to obtain results. He can then see a faulty condition as soon as it takes place or even starts to take place, and remedy it promptly. If such readings are recorded continuously the fireman can see how quickly the remedy is effective. Such records also show the executive in charge of the plant just what took place at all times.

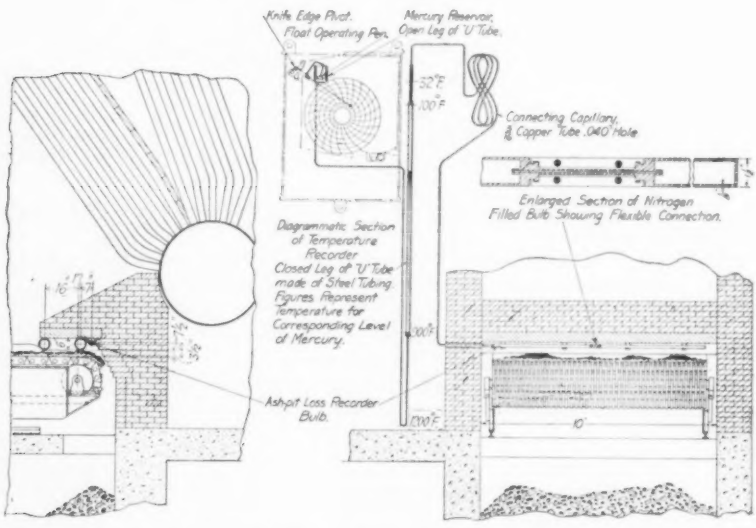


FIG. 1 ASH-PIT LOSS RECORDER INSTALLATION ON NATURAL-DRAFT CHAIN-GRATE STOKER IN STATION B

4 It is the purpose of this paper to present some preliminary data obtained from a device recently developed to record the ash-pit loss from chain-grate stokers. It may later be extended to some other types of stokers of either the present or modified design, but the great need for it on chain grates led to it being applied to these first.

METHOD OF RECORDING ASH-PIT LOSS

5 The fundamental principle used in recording the ash-pit loss by this method is to place a temperature-recorder bulb at a point near where the ashes are discharged from the active section of the grate or furnace. The bulb is not in contact with the ashes but receives only the radiant heat from the coke being discharged. The relation between this temperature and the amount of combustible

in the refuse, and the heat loss due to it, is established by taking a series of samples corresponding to different temperature readings.

6 If the ashes are well burned out, so that there are no live coals, the temperature is comparatively low. Even though the ash has clinkered rather badly and retained a certain amount of temperature, it has usually lost its glow and is comparatively cold from a radiation standpoint. The coke coming over is still ignited and at a temperature of about 1800 to 2000 deg. fahr., so that its radiant heat to the bulb is sufficient to cause very decided changes in the temperature readings.

7 In selecting a location for the temperature-recorder bulb, it is necessary to have it shielded from any radiant heat of the furnace itself, and also as free as possible from changes in temperature due to the surrounding brickwork, circulation of air, etc.

8 It is necessary that the bulb extend across the entire width of the grate and be so constructed as to obtain an accurate average in case the amount of coke coming over is greater on one portion of the grate than another.

9 Fig. 1 shows an installation on a natural-draft chain-grate stoker with a 9-ft. bulb located just beyond the water back, which shields it from the heat of the furnace. It is located about seven inches above the rear end of the grate, so that the radiant heat from any coke passing under it will increase its temperature.

THE TEMPERATURE RECORDER

10 The temperature-recorder bulb itself is made very rugged to withstand this service. It is constructed of $1\frac{1}{2}$-in. steel tubing with $\frac{1}{8}$-in. wall and all joints are welded. The bulb is charged with nitrogen and the gas pressure that is exerted by an increase in temperature is conducted through a $\frac{3}{16}$-in. copper capillary tube to the recorder. The recorder consists of a simple mercury U-tube, one leg of which is open to the atmosphere and carries a float to which the recorder pen is attached.

11 The other leg of the U-tube is made up of sections having different cross-sectional areas, so that the pen movement is suppressed below and above the range to be recorded on the chart. It is also possible, if necessary, to have the area of one leg of the U-tube variable, so that the percentage of heat loss will be recorded directly with uniform graduations on the chart in case the relationship between the ash-pit loss and temperature should not be a straight line.

12 The bulb can be made in sections short enough to be installed where the alley between two boilers is narrower than the width of the boiler itself. Each section, however, is welded permanently with ¼-in. No. 13 gage steel tube, making a flexible connection so that it can be folded up and then straightened out as it is installed in the furnace.

LOCATION OF TEMPERATURE-RECORDER BULB IN FURNACE

13 Up to the present time the experiments have been largely confined to the selection of the best locations for the bulb with respect to the grate and also to determine the relative advantage of bare bulbs and those having lagging around a portion of their

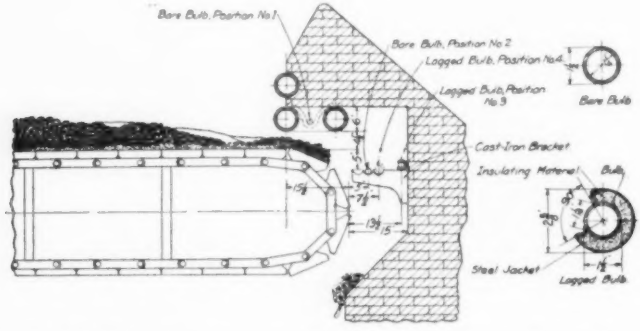


FIG. 2 ASH-PIT LOSS RECORDER ON FORCED-DRAFT CHAIN-GRATE STOKER IN STATION A, SHOWING DIFFERENT BULB CONSTRUCTIONS AND LOCATIONS

circumference so as to minimize the heat received from or given off to the surrounding walls, air, etc.

14 Fig. 2 shows the location of a bulb at the end of a forced-draft chain-grate stoker. A bare bulb was first located at position No. 1 with cast-iron supports. It was found that when the fuel was up to the bottom of the waterback and there was a slight positive pressure in the furnace, a flame would envelop the bulb, sending the temperature up to more than 1500 deg. No data were obtained from this recorder in the No. 1 position.

15 The bare bulb was next placed on the cast-iron bracket in position No. 2, and after operating several days it was found to be satisfactory as to temperature range and continuous operation. A series of calibration tests was then started to learn what combustible losses corresponded to the different temperature readings.

METHOD OF SAMPLING ASHES

16 Samples consisting of *all* the ash discharged from the stoker were taken hourly. The ash as it accumulated in the hopper was quenched immediately by means of a water spray. At the end of the hour the ash was dumped from the hopper, crushed, mixed and quartered by hand until it was reduced to about 20 lb. of 4-mesh size, and sent to the laboratory. The analyses show combustible on the dry basis.

TABLE 1 ASH-PIT LOSS-RECORDER TEST DATA FROM STATION A

(Results from two days' tests)

Date	Time	Boiler Rating, per cent		Ratio of air flow to steam flow	Total air	Ash-pit loss rec temp., deg. fahr.	Combustion refuse, per cent	Heat loss, per cent	Grate speed, in. per min.	Draft over fire, inches water	Flue-gas temp. deg. fahr.
		Steam flow	Air flow								
2/10/21	9:00-10:00	170	177	1.04	146	421	21.84	4.50	2.31	0.07	614
(Coal,	10:00-11:00	173	176	1.02	143	590	29.98	7.00	3.76	0.06	616
10.6	11:00-12:00	173	177	1.02	143	612	29.60	6.80	3.48	0.06	620
per	12:00- 1:00	175	177	1.01	141	595	28.56	6.50	4.12	0.07	621
cent	1:00- 2:00	176	179	1.01	142	597	28.68	6.60	3.97	0.07	620
ash)	2:00- 3:00	174	180	1.03	145	601	30.74	7.30	4.12	0.09	631
	3:00- 4:00	176	180	1.02	143	622	28.14	6.40	3.97	0.10	623
3/2/21	11:15-12:15	153	152	0.995	139	571	38.15	5.65	3.02	0.02	580
(Coal,	12:15- 1:15	149	151	1.01	142	555	35.50	5.04	3.20	0.02	580
8.4	1:15- 2:15	148	145	0.98	137	564	40.28	6.20	2.66	0.01	580
per	2:15- 3:15	105	110	1.05	147	463	36.28	5.23	2.13	0.01	539
cent	3:15- 4:15	126	134	1.06	149	450	34.66	4.87	2.48	0.01	558
ash)	4:15- 5:15	134	176	1.31	184	478	32.88	4.49	2.84	0.03	591

17 The samples from Station A were collected from time to time between January 20 and March 2, 1921, at times when it was convenient to do the work. Usually five to eight samples were taken each day. In all, 107 samples were taken during 15 different days. About half of them were taken during certain stoker tests that were being made on this boiler, and the remainder under every-day operating conditions.

18 No special effort was made to change operating conditions from normal except in the case of a few samples, when more combustible was run over than usual in order to get some high points to establish the upper part of the curve.

RELATION BETWEEN TEMPERATURE AND COMBUSTIBLE LOSS

19 Table 1 gives a sample of the data obtained and averaged for each hour, representing the conditions existing while taking the 107 samples. The essential results are plotted in Fig. 3, where each small circle represents the average of all temperature readings grouped between 300 and 400, 400 and 500 deg. fahr., etc. It is noted that

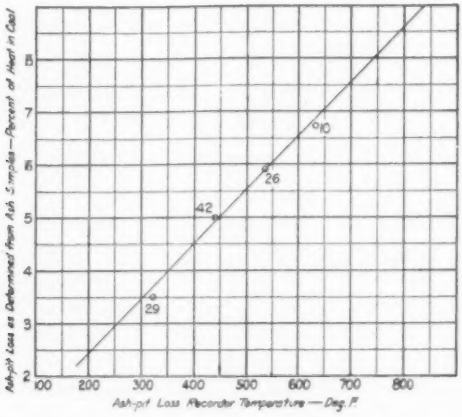


FIG. 3 ASH-PIT LOSS AND TEMPERATURE RELATION FOR BARE BULB IN POSITION NO. 2 (FIG. 2) ON FORCED-DRAFT CHAIN-GRATE STOKER AS OPERATED IN STATION A

(Figures represent number of individual tests averaged.)

the ash-pit loss is about 3.5 per cent at 300 deg. and increases 1 per cent for each 100 deg. temperature rise.

20 It was first thought best to plot the results between temperature and percentage of combustible in the ashes, but it was found that the ash in the coal varied between 8.5 and 14.0 per cent. From the curves of Fig. 4 it is noted that with samples having 30 per cent combustible the heat loss is 3.9 per cent from the 8.5 per cent ash coal and 7.0 per cent from coal with 14 per cent ash, hence each test was figured on a basis of the heat loss due to unburned carbon, expressed in percentage of the heat in the coal.

21 The relations between percentage of combustible in refuse, ash in coal, and ash-pit loss are given in the series of curves in Fig. 4. These curves are plotted from the formula:

$$\text{Combustible loss} = \frac{100\,ac}{(100-a)\,(100-c)}$$

where a is the percentage of ash in the coal and c the percentage of combustible in the refuse. This formula is based upon the assumption that all of the ash from the coal goes to the ash pit and that the heating value per pound of combustible in the ash pit is the same as the combustible in the coal.

22 Emphasis is usually placed upon the percentage of combustible in the ashes as being the true measure of loss due to unburned carbon, but it is of equal importance to know the percentage of ash in the coal. There is more heat lost with 20 per cent combustible in

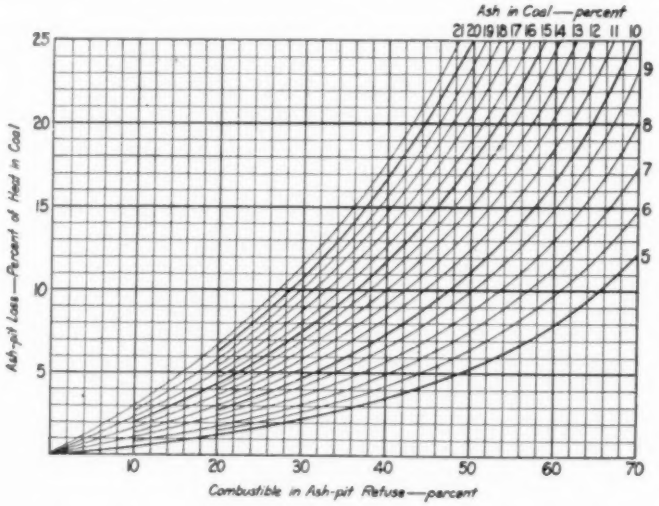


FIG. 4 RELATION BETWEEN ASH-PIT LOSS, COMBUSTIBLE IN REFUSE, AND ASH IN COAL

the refuse from a coal with 20 per cent ash than with 40 per cent combustible in the refuse from a coal with 8 per cent ash.

TEMPERATURE VARIES WITH PERCENTAGE OF HEAT LOST IN COMBUSTIBLE

23 One naturally questions whether the temperature of a bulb at such a location will vary with the percentage of heat lost or with the percentage of combustible in the refuse; whether either relation will hold true for different percentages of ash in the coal, or whether it varies with the boiler rating, grate speed, etc. The final answer to such questions can only be determined from results of actual tests.

Fig. 3 is plotted from the 107 samples and it shows conclusively that the temperature varies directly with the percentage of the heat of the coal lost in unburned carbon. Various other plots and studies of the individual tests fail to show any tendency for the results to deviate from this curve due to changes in the percentage of ash in the coal, the grate speed, or the boiler rating.

24 Radiant heat absorbed by a given object varies with the temperature, distance, and area of the radiating body. The radiating body in this case is the live coke at the upper part of the curvature at the end of the grate. The distance from the bulb to the point where the coke and ash break and fall from the grate is practically constant. The temperature of the coke is practically constant, as its glow is maintained by slow combustion with free air at this point. The temperature of the mass of coke itself tends to increase somewhat with the amount of coke, and perhaps slightly with the grate speed.

25 The exposed area of the heat-radiating combustible varies almost directly with the average thickness of the layer of coke on the moving grate, for as the coke breaks it leaves a fresh cross-section exposed. The effect is the same as if the grate were standing still; the exposed area changes like a moving picture on the screen, with the bulb as the spectator.

26 As the speed of the grate is increased the actual weight of coke lost increases for a given cross-sectional area and temperature reading, but the rate of coal feed has also increased substantially in proportion to the grate speed, thereby leaving the loss of coke a constant percentage of the total coal and therefore still proportional to the temperature reading.

27 Any increase in the percentage of ash in the coal has the effect of diluting the combustible in the ashes by just that amount. To carry the same load on the boiler will require the same actual weight of combustible in the coal, and therefore an increased in the amount of coal fed in proportion to the increase in the ash, or slightly greater if the larger amount of ash decreases the efficiency. If the actual amount of the coke coming over the end of the grate is the same, the temperature of the bulb at position No. 2, Fig. 2, will remain the same, even though the percentage of combustible in the sample of refuse is lower because of its dilution by the larger amount of ash. It would therefore appear that the percentage of ash would produce no direct or indirect effect upon the relation between the amount of combustible lost and the temperature of the bulb. It is

therefore evident that the temperature varies with the percentage of heat lost, and not with the ratio of combustible to ash in the sample of refuse.

28 This is very fortunate as the temperature record can then be used directly in determining the heat loss without the necessity of bothering with percentage of ash in coal or any other modifying factors. In fact, the chart can be graduated direct in percentage of heat lost instead of temperature.

29 Samples of coal and combustible are needed only for making the calibration tests, which should be determined for each type of installation. At least this will be necessary until more data have been obtained regarding location of bulb and the characteristics of different types of stokers, arrangements of water backs, etc., as these are likely to produce a different rate of temperature change, as shown later in this paper.

PERFORMANCE RESULTS FROM CHAIN-GRATE STOKERS

30 Fig. 5 is a chart record, reduced in size, which shows certain changes in the relationship between the various factors as the stoker operating conditions change. For instance, the average ash-pit loss is greater when the air-flow and steam-flow pens are together than it is when the air flow reads more than the steam flow. In this case these two pens are together when the excess air is 40 per cent, and the total air increases with the ratio of air flow to steam flow. During the night when the load is light the air flow was permitted to continue at about 180 per cent of boiler rating while the steam flow went down to 80 or 90 per cent, corresponding to a total air of 300 per cent or 200 per cent in excess of that required for combustion. During this time the ash-pit loss recorder went to the stop which corresponded to 300 deg. for the bare bulb and a temperature of 300 to 350 deg. for the lagged bulb.

31 It is well known that with a chain-grate stoker the ash-pit loss tends to increase as the excess air is decreased, and vice versa. It is only natural that more coal should be wasted over the end of the grate when the fuel bed is kept thicker and carried to the water back. When the fire burns short a reduction is made in the ash-pit loss, but there is a decided increase in the percentage of excess air. As a matter of fact this is generally true in all types of stokers, and even in hand firing. That is, a reduction in unburned coal usually means some increase in excess air, so that decreasing one loss increases another.

32 The problem is then to determine what method of operation will give the lowest total loss, and the ash-pit loss recorder makes it possible to plot out these relations as combustion characteristics of the stoker.

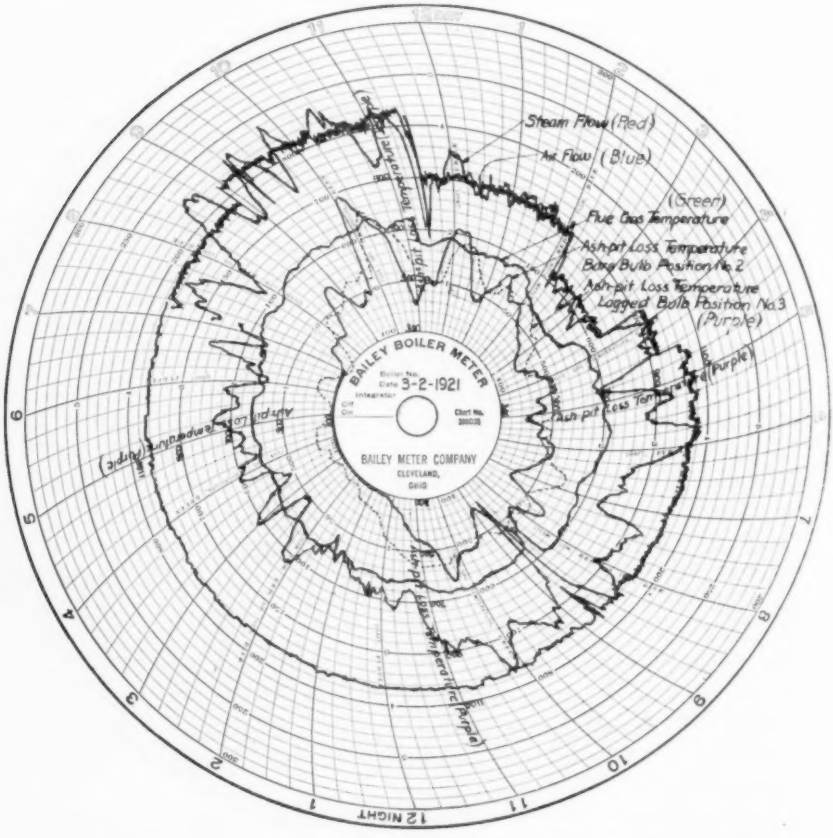


FIG. 5 CHART FROM ASH-PIT LOSS RECORDER IN STATION A

RELATION BETWEEN ASH-PIT LOSS AND TOTAL AIR USED FOR COMBUSTION

33 Fig. 6 shows a very definite relation between temperature as indicated by the ash-pit loss recorder and the total air used for combustion on a basis of the theoretical air being 100 per cent. In

plotting the points the total air was taken from the ratio of air flow to steam flow as shown on chart records similar to that of Fig. 5, using only periods when the steam flow, the air flow and the ash-pit loss-recorder temperature were each very uniform over stretches of two hours or more at one time, indicating that the conditions had settled to normal and were not varying continuously as on the chart of Fig. 5. At no time on the latter chart were the conditions uniform enough with respect to these factors to use the results in plotting Fig. 6.

34 The values taken from the average curve of Fig. 6 are used to calculate the heat losses and are plotted in Fig. 7, showing the characteristic loss curves for a forced-draft chain-grate stoker as operated in Station A. This set of curves shows that the percentage of heat lost in excess air (based on a 500-deg. temperature rise) increases from zero up to 19 per cent at 300 per cent total air. As the excess-air loss is reduced the ash-pit loss increases, but the rate of increase is less than the rate of decrease in the excess-air loss, so the total of these losses decreases until the total air is about 130 per cent. At this point CO and unburned gases appear and the percentage of heat lost in consequence increases very rapidly as the amount of air is reduced to 100 per cent, or the theoretical amount. It is therefore best to operate this stoker at about 130 to 140 per cent total air.

35 It is quite possible that, with a little better care upon the part of the fireman, the combustible loss could be considerably reduced on this type of stoker, even at this low percentage of excess air. This would then lower the total curve so that the combined loss would be still lower than the 10 per cent shown in Fig. 7 as the minimum point.

RELATION BETWEEN CAPACITY AND STOKER LOSSES

36 Fig. 8 shows the relation between the boiler rating and flue-gas temperature, excess air, unburned combustible and CO, with the percentage of heat lost in each. The total of these three losses is seen to be at a minimum of 11 per cent at 225 per cent rating and to remain below 12 per cent at all ratings between 160 and 260 per cent.

37 The minimum loss in Fig. 7 is 10 per cent because the excess-air loss is calculated on the arbitrary basis of 500 deg. fahr. rise in flue-gas temperature, while in Fig. 8 it is calculated on the actual temperature, which is as high as 650 deg. at the higher rating.

38 If it were possible to operate this stoker with lower excess air at the lower ratings and lower ash-pit loss at the higher ratings, the total losses would be materially decreased and the curve would be flatter. In fact, the range of best efficiency as shown in Fig. 7 should be almost entirely independent of boiler rating if the same care is exercised by the fireman at the low ratings as he must exercise at the high ratings to produce the steam. The chart record, Fig. 5, shows careless operation when the dampers are left open and the same amount of air is used during the light-load period at night as is used

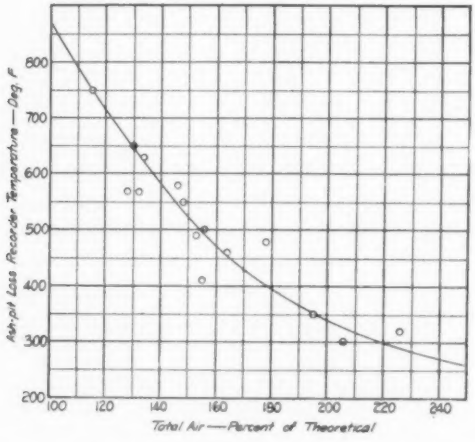


FIG. 6 RELATION BETWEEN TOTAL AIR USED FOR COMBUSTION AND ASH-PIT LOSS-RECORDER TEMPERATURE OBTAINED FROM FORCED-DRAFT CHAIN-GRATE STOKER OPERATED IN STATION A

during the high load peaks. The period between 2.15 and 3.45 p.m. shows that the air flow can be reduced with the steam flow and maintain a low total loss. The average conditions during this 1½-hour period show 105 per cent boiler rating, 147 per cent total air, 540 deg. flue-gas temperature and 340 deg. ash-pit loss-recorder temperature. These correspond to 4.4 per cent excess-air loss, 4.0 per cent ash-pit loss, and no CO, hence a total loss of 8.4 per cent, while the results from Fig. 8 show a loss of 18 per cent at 110 per cent rating under average operating conditions.

39 To obtain the total overall efficiency curves for stoker, furnace, boiler and perhaps economizer, the other losses due to products of combustion and infiltration of air should be included according

to the equipment used. There is also the question of overhead charges and repairs to stokers and brickwork that increase with higher rating or lower excess air.

CHARACTERISTIC CURVES OF STOKERS

40 This set of curves raises the question, Can true characteristic curves be drawn showing the performance of the stoker itself? Very little has been done in determining the performance characteristics of stokers. Such data have been well established for centrifugal pumps, motors, turbines, fans and blowers, but in the operation of a

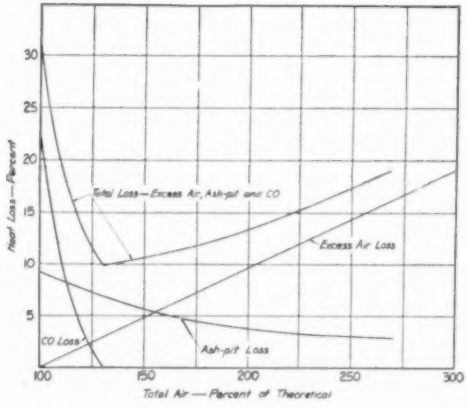


FIG. 7 CHARACTERISTIC LOSS CURVES, ON BASIS OF TOTAL AIR, OF FORCED-DRAFT CHAIN-GRATE STOKER AS OPERATED IN STATION A

stoker so much depends upon the character and quantity of coal being burned and also upon the human element of the fireman, that the true characteristics of the stoker are usually so clouded that they do not appear in their true form.

41 Most boiler tests give combined efficiency of furnace and boiler as it is difficult to separate the two. In reality it is not necessary to separate these two efficiencies or even to determine the efficiencies at all, because a great deal more can be accomplished through studying the losses than can be obtained from knowing the overall efficiency alone. We know that if the losses are high the efficiency is low, and vice versa. It is also easy to separate certain of the losses and charge them directly to the furnace, while other

losses may be chargeable entirely to the boiler and still others are beyond the control of either.

42 The three important losses existing in boiler and furnace operation that are chargeable directly to the stoker and furnace are unburned combustible in ashes, unburned gas, and excess air. The latter two losses may be influenced somewhat by the size of the furnace and combustion chamber, and the last one by the leakage of air through the boiler setting, but on the whole they depend almost entirely upon the stoker and its operation.

43 If the characteristics of the stoker are properly determined,

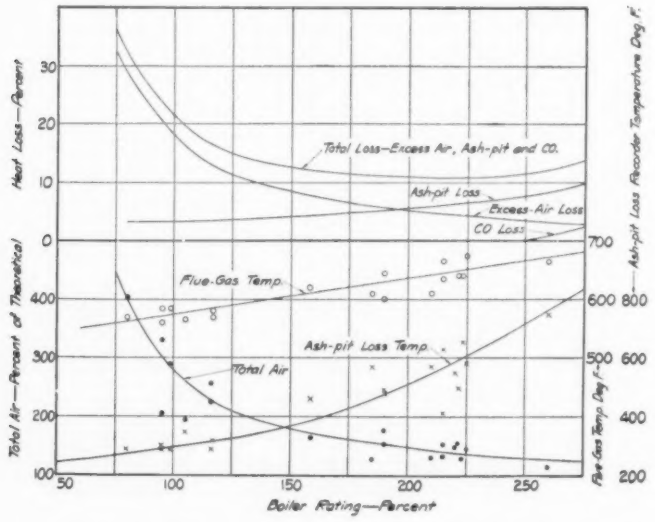


FIG. 8 CHARACTERISTIC LOSS CURVES, ON BASIS OF BOILER RATING, OF FORCED-DRAFT CHAIN-GRATE STOKER AS OPERATED IN STATION A

one can then see the best relation between these factors in order that the total of the three losses be a minimum. We are then assured the furnace efficiency is highest, even though we may not know the exact thermal efficiency of the furnace and stoker itself, separate from the boiler.

44 In operating stokers it has been considered that the character of the coal and the method of operation were far more important than the characteristics of the stoker itself. This, however, is only an admission that the stoker was not so designed that its character-

istics would dominate over the other factors, namely, coal and fireman. Very great improvements have been made in stoker design and methods of operation, but there is need for still further advancement. A careful analysis of the problem by means of continuous graphic records of all the losses and characteristic performance curves like those shown in Figs. 7 and 8 will aid in further improvements

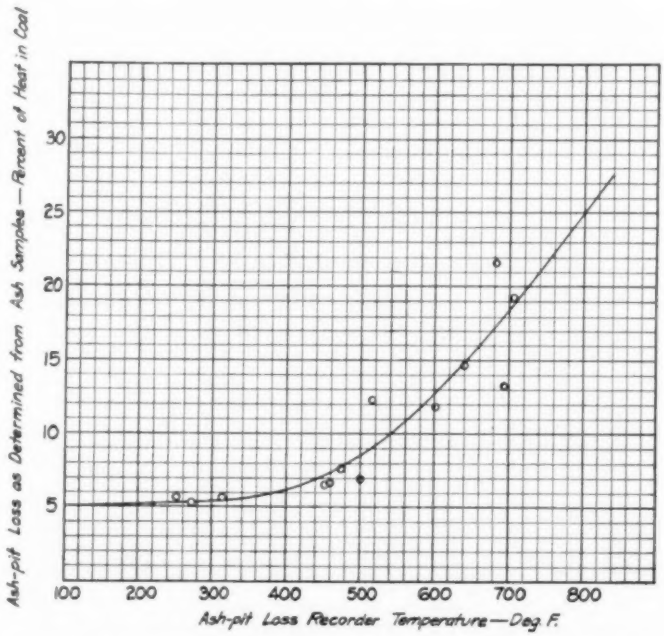


FIG. 9 ASH-PIT LOSS AND TEMPERATURE RELATION. BULB AS IN FIG. 1 ON NATURAL-DRAFT CHAIN-GRATE STOKER, STATION B

(Each point represents an individual one-hour sample.)

in both design and operation, as well as in selecting coal for the stokers or the selection of a stoker for the coal available.

45 Even where ash-pit loss recorders cannot now be easily applied to some types of stokers, it is possible to take hourly samples of the refuse and plot characteristic combustion and capacity curves like those in Figs. 7 and 8 For stokers with intermittent ash dumps the period for each test should be the same as the period between dumps, maintaining the operating conditions as uniform as possible during this period and taking the ash sample from each dump. As

the value of these results does not depend upon coal weights and evaporation efficiencies, short tests are satisfactory for determining the losses due to excess air, unburned gas, and unburned coal. All of the losses of the boiler heat balance, except radiation, can be determined almost, if not quite, as accurately as can be the evaporation.

RESULTS FROM NATURAL-DRAFT CHAIN-GRATE STOKER AT STATION B

46 Table 2 and Figs. 9 to 13 give all of the data from thirteen ash samples and tests on a natural-draft chain-grate stoker equipped

TABLE 2 ASH-PIT LOSS-RECORDER TEST DATA FROM STATION B[1]

(Coal, 11.78 per cent ash; 12,732 B.t.u.)

Date	Time	Boiler Rating, per cent		Ratio of air flow to steam flow	Total air	Ash-pit loss-rec. temp., deg. fahr.	Combustion refuse, per cent	Heat loss, per cent	Grate speed, in. per min.	Draft over fire, inches water
		Steam flow	Air flow							
1921										
2/24	3:30- 4:30	56	105	1.87	303	500	34.10	6.95		0.15
	4:30- 5:30	95	116	1.22	197	518	47.85	12.25		0
	5:30- 6:30	107	116	1.08	175	602	46.80	11.80	5.00	0.17
	6:30- 7:30	96	122	1.27	206	683	61.85	21.60	5.25	.15
2/25	10:45-11:45	101	123	1.22	197	640	52.05	14.60	4.62	
	11:45-12:45	105	113	1.07	173	707	58.90	19.20	5.22	
	4:00- 5:00	68	120	1.77	287	475	36.00	7.55	2.80	
	5:00- 6:00	107	117	1.09	176	695	49.70	13.22	5.20	
2/26	8:00- 9:00	76	98	1.29	209	247	29.65	5.68	4.00	
	9:00-10:00	60	112	1.87	303	272	27.70	5.13	2.50	
	10:00-11:00	71	95	1.33	216	315	29.50	5.60	3.84	
	11:00-12:00	89	91	1.02	165	454	32.35	6.41	3.85	
	12:00- 1:00	71	84	1.20	194	460	33.15	6.68	3.97	

[1]Results of three days' tests.

with an ash-pit loss recorder as shown in Fig. 1. These data were obtained on three consecutive days from a boiler that was being operated in a stand-by station. The load conditions did not require careful attention to the stoker operation, hence the results should not be taken as indicative of the best performance with this type of stoker. Neither should these results be used for comparing one type of stoker with another, because the operating factors, namely, coal and fireman, may be very different.

47 In Fig. 9 the ash-pit loss increases very slowly with the initial rise in temperature, and then more rapidly at the higher temperatures. Referring to Fig. 1, it will be seen that the bulb is located at a different position with respect to the fuel bed from the one in Station A. In this case the bulb is directly over the end of the grate, and looks down upon the coke as it passes beneath it. With a very short fire the coke may "go out" or cease to burn before it reaches the water back and yet contain a considerable percentage

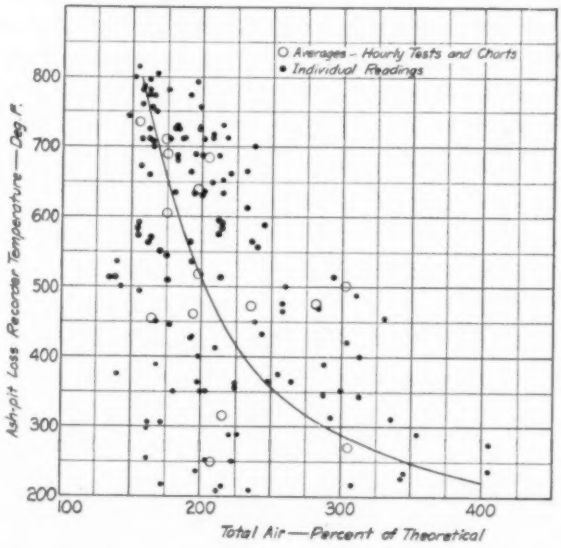


FIG. 10 RELATION BETWEEN TOTAL AIR USED FOR COMBUSTION AND ASH-PIT LOSS-RECORDER TEMPERATURE OBTAINED FROM NATURAL-DRAFT CHAIN-GRATE STOKER AS OPERATED IN STATION B

of combustible, hence the minimum loss of 5 per cent with little or no radiant heat. As soon as a few pieces of hot coke are carried over the effect upon the bulb is very marked, and a considerable rise in temperature takes place until the grate is covered with coke. A greater amount of combustible will add only to the thickness of the layer and not increase the area of the hot coke within the vision of the bulb in this location. The increase in the temperature of the bulb is due only to the closer proximity of the hot coke to it after the grate is once covered.

48 In Fig. 10 the relation plotted between total air and ash-pit

loss-recorder temperature shows widely scattered points. This is largely because the fuel-bed conditions vary rapidly, and there is considerable time lag between the condition of the fuel bed on the grate proper and the amount of coke being discharged at the same time. The log, Fig. 13, plotted over a short period, shows a typical case of the variations as the fire burns thin and is again thickened. The ash-pit loss-recorder temperature drops with the short fire, but the lowest temperature is not reached until twenty minutes after the greatest air excess and the lowest steam flow.

49 The conditions of the fuel bed on this stoker are sometimes

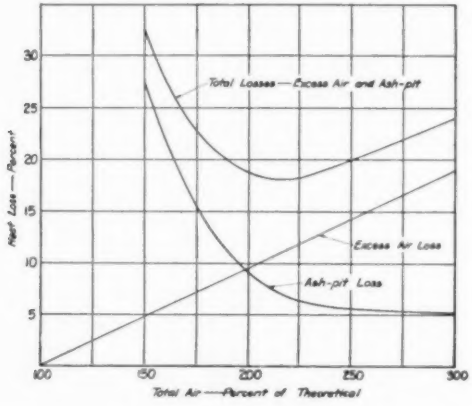


FIG. 11 CHARACTERISTIC LOSS CURVES, ON BASIS OF TOTAL AIR, OF NATURAL-DRAFT CHAIN-GRATE STOKER AS OPERATED IN STATION B

very good and sometimes very poor, just as it may happen, due either to the stoker characteristics, the human factor, or the coal factor, or to a combination of all three. In any case there are certain average conditions which tend to repeat and show the relationship between the various factors. It is very evident from Fig. 10 that very high excess-air and high ash-pit losses do not occur at the same time, although either may be low when the other is high; but they should both be low for the best efficiency.

RELATION BETWEEN ASH-PIT AND TOTAL AIR

50 In Fig. 11 it will be noted that the natural-draft chain-grate stoker in question, as it was being operated at this time, gives the best efficiency at about 210 per cent total air, with a loss of 18

per cent due to excess air and unburned coal. We all know that better results than this can be obtained under suitable load conditions with good operation and the right kind of coal, yet these curves indicate that there is a tendency for the losses to increase very rapidly as the fire changes to a shorter or longer condition from the most economical point.

51 The air having free and equal access to all parts of the grate, takes the path of least resistance through the fuel bed and develops holes very rapidly when once started. In many plants the boilers are operated with the damper wide open. If the maximum amount of steam is not needed. the fire burns short. If more steam is needed, the stoker is speeded up and a little greater proportion of the grate area is covered with active fuel bed. In other words, the steam pressure is regulated by means of excess air. Figs. 5 and 13 bring out this point very nicely, as the air flow remains practically constant and more or less steam is produced as the coal is fed into the path of the air.

CHARACTERISTIC CURVES FOR NATURAL DRAFT

52 Fig. 12 shows the excess-air and ash-pit losses plotted on a basis of boiler rating. The total of these two losses has its lowest value (18 per cent) at about 90 per cent rating. At lower ratings the excess-air loss increases rapidly and at higher ratings the unburned-coal loss increases even more rapidly.

53 One naturally wonders whether the ash-pit loss would continue increasing at the rate indicated by the slope of this curve if the boiler were operated at still higher rating. The limit in this direction is reached when the space under the water back is completely filled with ash and coke The percentage of combustible lost when operating with the space under the water back completely filled with ash and coke combined, will depend upon the percentage of ash in the coal, the coal-gate opening and the height of the water back above the grate, and will be practically independent of the speed of the grate.

54 If the increased rate of combustion is obtained by increasing the gate opening and the intensity of draft, leaving the grate speed and the space under the water back constant for a given percentage of ash in the coal, a decrease in the ash-pit loss will result as shown by the upper dotted curve at the right of Fig. 12, because there will

then be more ash per square foot of grate and therefore less space left for coke under the water back.

55 For any given relation between gate openings and space under water back the percentage of combustible lost would decrease with increased percentages of ash in the coal and should of course remain constant for different ratings. A change in the height of the water back would also have a similar effect the percentage of ash-pit loss remaining the same for all ratings in both cases so long as the space under the water back was kept completely filled and increased ratings obtained by increasing grate speed.

56 If the fire burns thinner the ash-pit loss will be decreased

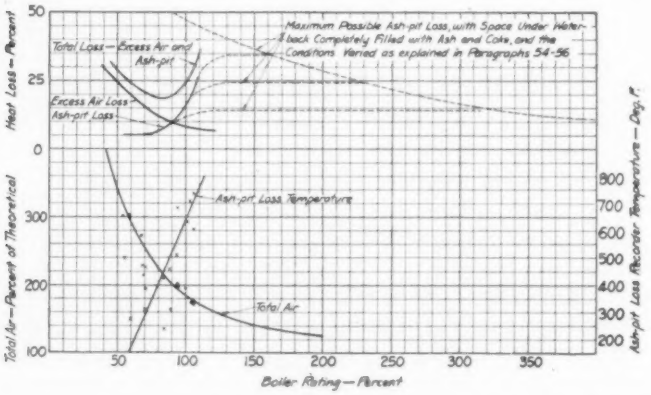


FIG. 12 CHARACTERISTIC LOSS CURVES, ON BASIS OF BOILER RATING, OF NATURAL-DRAFT CHAIN-GRATE STOKER AS OPERATED IN STATION B

for any of the foregoing conditions and the excess-air loss will naturally increase.

57 It is believed that characteristic curves like those shown in Figs. 11 and 12, plotted for different relations between the factors that have been mentioned, will be very helpful in obtaining the best results from natural-draft chain-grate stokers. By means of such curves we should be able to accomplish better results in the arrangement of water backs regulation of coal gate grate speed. draft, etc., and also assist the fireman in obtaining the required amount of steam with the minimum loss.

CONCLUSIONS

58 From a study of the data obtained the following conclusions may be drawn:

a The temperature of a bulb properly located near the rear of a chain-grate stoker responds definitely to changes in the amount of combustible going to the ash pit.

b The changes in temperature vary in direct proportion to the percentage of heat lost to the ash pit when the bulb is properly located.

c This relation between temperature and ash-pit loss is not

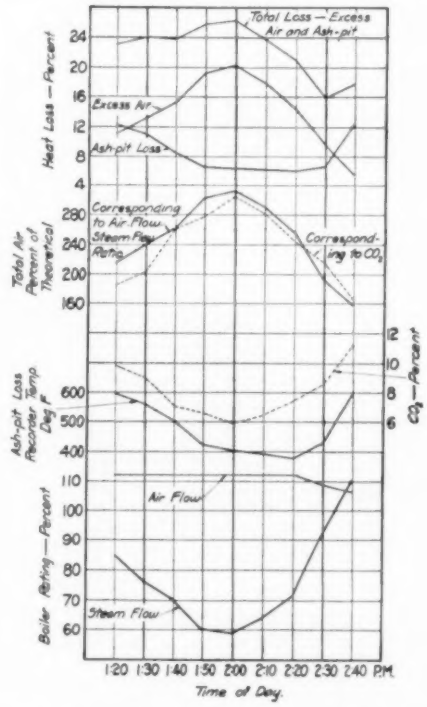


FIG. 13 LOG SHOWING CHANGE OF FUEL BED ON NATURAL-DRAFT CHAIN-GRATE STOKER AS OPERATED IN STATION B

materially affected by grate speed, rate of combustion, or percentage of ash in coal, but should be determined for different types of installations by actual tests.

d A continuous record of the ash-pit loss enables the fireman to effectively control this important factor.

e Records of ash-pit loss, excess air, unburned gas, flue-gas temperature, and boiler rating make it possible to plot characteristic

performance curves for stokers showing the losses due to unburned combustible, unburned gases, and excess air, and to determine the proper amount of excess air to be maintained and also the most efficient capacity.

f These stoker characteristic curves can be used to compare the relative efficiencies of different types of stokers, the suitability of different kinds of coal, and the effectiveness of the control by the fireman.

SUPPLEMENTAL DATA, STATION C

59 Fig. 14 shows the location of a bare bulb position No. 1 immediately behind the water back supported on the cast-iron members which carry the fire brick, protecting the I-beams.

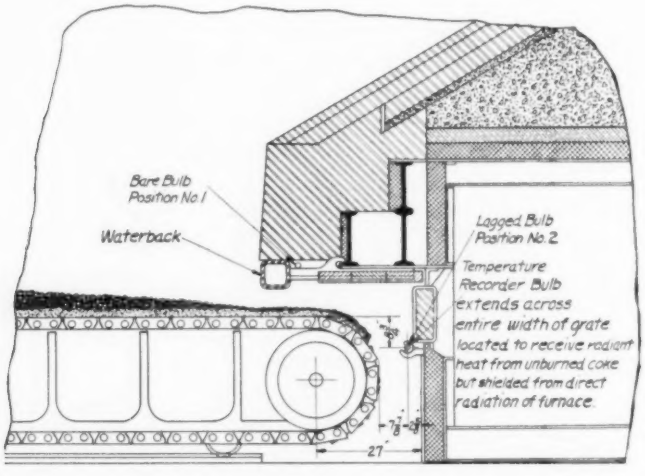


FIG. 14 ASH-PIT LOSS RECORDER INSTALLATION ON FORCED-DRAFT CHAIN-GRATE STOKER IN STATION C

60 Another bulb is shown in position No. 2, this being lagged and supported from cast-iron members projecting into the rear wall. This installation is on a 1400-hp. boiler having two forced-draft chain-grate stokers with a partition wall between. The bulbs as shown are on one stoker only, although the other stoker on this boiler was equipped with a bulb corresponding to position No. 1 and later was supplemented with a lagged bulb corresponding to position No. 2, as this was found to be more reliable in the results obtained.

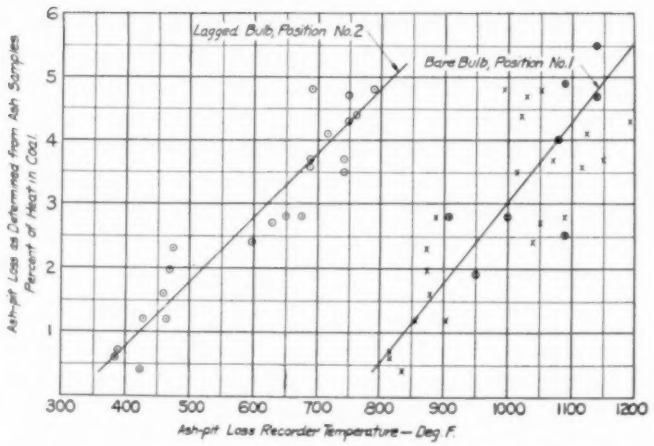


FIG. 15. ASH-PIT LOSS AND TEMPERATURE RELATIONS FOR BARE AND LAGGED BULBS IN POSITION NOS. 1 AND 2, FORCED-DRAFT CHAIN-GRATE STOKER STATION C.

⊙ Individual one-hour samples, lagged bulb × Individual one-hour samples, bare bulb
Boiler test samples, bare bulb

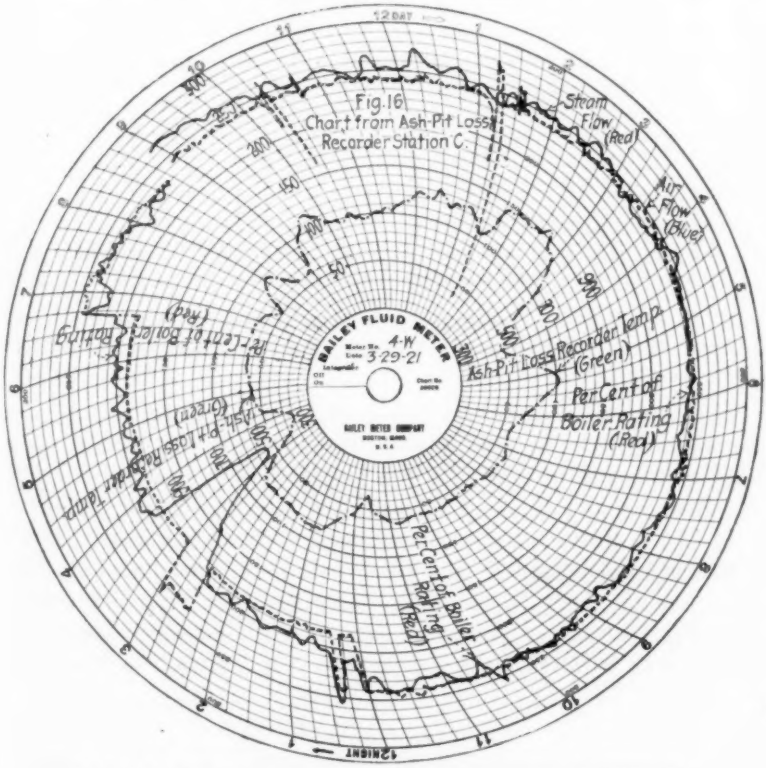


FIG. 16 CHART FROM ASH-PIT LOSS RECORDER, STATION C

61 Fig. 15 gives data from a large number of one-hour samples, where the readings of the two temperature recorders were taken and, of coursé, correspond to the same individual sample of refuse.

62 Some additional information is also shown covering several boiler test samples which were made before the lagged bulb was installed in position No. 2 and, therefore, apply only to the bare bulb in position No. 1. It will be noted that the temperature of

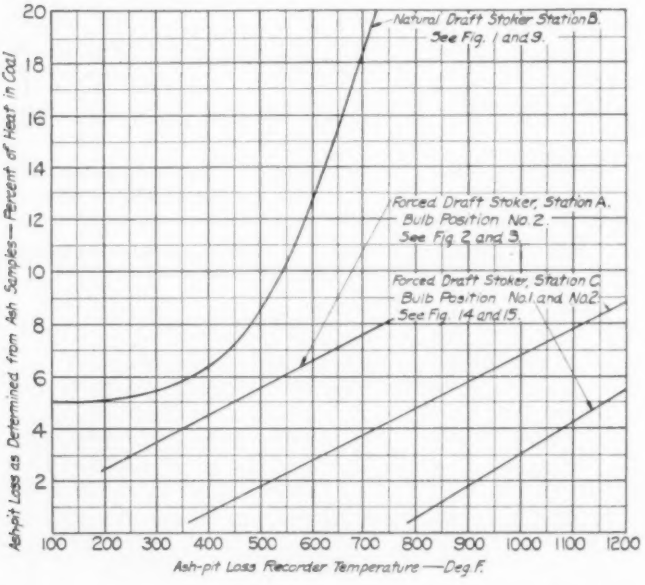


FIG. 17 COMPARISON OF RELATION BETWEEN ASH-PIT LOSS AND TEMPERATURE AS DETERMINED IN DIFFERENT STATIONS AND BULB POSITIONS

position No. 1 is about 400 deg. higher than position No. 2, both curves having about the same inclination but the results seem slightly more consistent from the lagged bulb in position No. 2.

63 Fig. 16 is a chart record taken from the lagged bulb position No. 2 during some of these tests. It will be noted that the temperature reached 900 deg. which would correspond to a combustible loss of about 5½ per cent. This does not represent normal operation because an excessive amount of coke was passed over temporarily in order to get some data. In normal operation, the

percentage of heat loss, due to combustible in the ash, is usually maintained below 2 per cent.

64 At the time of presenting this paper, no data were available from both stokers to plot the characteristic curves showing relation between the excess air and combustible loss.

65 Fig. 17 is a composite plot of data obtained from Stations A, B, and C, showing the curves from bulbs located at the end of the stoker in Stations A and C are practically parallel, increasing approximately 100 deg. in temperature for for each one per cent of heat lost to the ashpit.

No. 1786

BOILER TESTS WITH PULVERIZED COAL

By Henry Kreisinger, Milwaukee, Wis.

and

John Blizard, Pittsburgh, Pa.

Members of the Society

In this paper the author presents a summary of the results obtained in an extended series of tests of a 468-hp. water-tube boiler fired with pulverized Illinois coal.

The results show that, contrary to the customary specification, it is not necessary to pulverize the coal to the extreme fineness of 85 per cent through a 200-mesh screen in order to get good combustion and good efficiency. This ability to burn coarser coal means increased capacity of the pulverizing mills and decreased cost of coal preparation.

The results also show that it is not necessary to dry coal to about 1 per cent moisture in order to burn it successfully in pulverized form, and it is stated that with most eastern coals drying is not at all necessary.

Good results can be obtained when the coal is burned at rates varying from $\frac{1}{2}$ lb. to 2 lb. per cu. ft. of combustion space per hour, and the best results at a rate of 1 to $1\frac{1}{2}$ lb.

THIS paper gives the summary of the results of a series of 11 tests made on a 468-hp. Edgemoor boiler equipped with a Foster superheater and fired with pulverized coal, at the Oneida Street Station of The Milwaukee Electric Railway and Light Company, Milwaukee, Wis. The tests were made by the Fuel Section of the U. S. Bureau of Mines in cooperation with the Research Department of the Combustion Engineering Corporation. The powdered-coal equipment was designed and installed by the Locomotive Pulverized Fuel Company. The coal burned in these tests came from the Illinois coal field. The object of the tests was to determine what overall efficiency can be obtained with pulverized Illinois coal under various conditions of furnace operation and different preparation of coal as to degree of fineness and percentage of moisture.

2 The tests were made in a thorough manner, everything being done to make the results accurate and reliable. The pulverized coal was weighed in specially designed tanks placed on platform

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scales as it was supplied to the furnace. The tests were of 17 to 25 hours' duration.

3 Tests Nos. 28 to 30, inclusive, were made with the usual preparation of coal as it is burned in the plant under ordinary operating conditions. Test No. 31 was made with the same condition of coal as in the three previous tests, but with the furnace provided with a cooling coil over the hearth and along the walls near the bottom

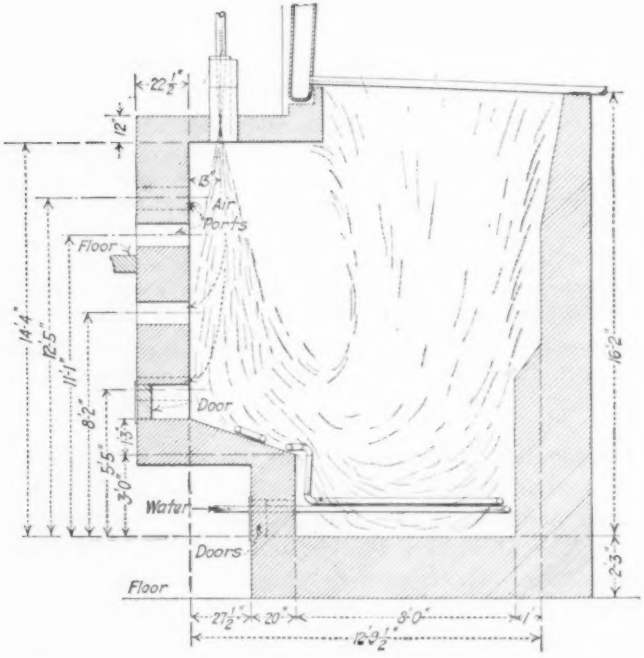


FIG. 1 SECTION THROUGH FURNACE, SHOWING ARRANGEMENT OF BURNER AND COOLING COIL OVER HEARTH AND NEAR BOTTOM OF FURNACE

of the furnace to facilitate the removal of ash. Tests Nos. 32 to 35, inclusive, were made with the same furnace arrangement as test No. 31, but with the coal pulverized to a lesser degree of fineness. Tests Nos. 36 to 38, inclusive, were made with the same furnace arrangement as in the previous four tests, but with undried coal.

4 Fig. 1 shows the arrangement of the burner and the cooling coil over the hearth and near the bottom of the furnace. The cooling coil consisted of three lengths of 2-in. pipe over the hearth and two

lengths along the side walls and the rear wall. The total surface of the coil was 48 sq. ft.

5 Fig. 2 shows the coal-weighing apparatus, which was placed between the storage bin and the feed bin. There were two burners and two feeders and the coal to each feeder was weighed separately. The weighing tanks were connected to the storage bins and the feeder bin by flexible canvas connections to permit weighing and to prevent the coal dust from escaping into the room when the weighing tanks

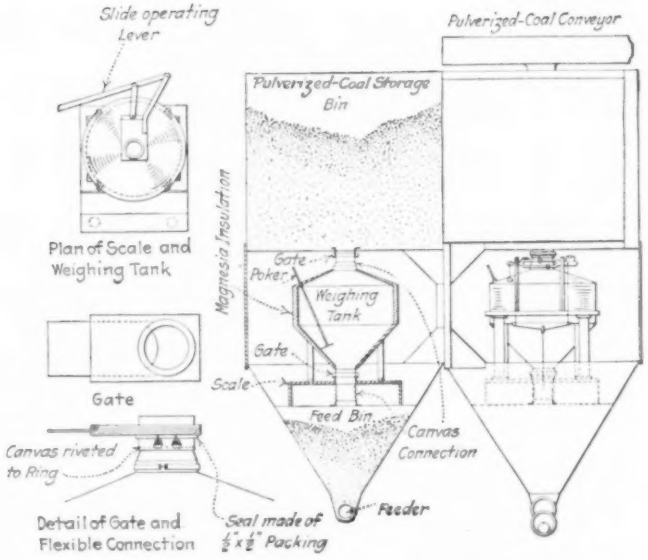


FIG. 2 COAL-WEIGHING APPARATUS USED IN TESTS

were filled and emptied. The tests were started and closed with the feeder bins empty.

6 The feedwater was weighed in two water tanks placed on platform scales. The water supplied to the cooling coil was measured by a 2-in. water meter which was calibrated at the rate of feeding the water through the cooling coil, and its measurements were found reliable to within less than one-half of one per cent.

7 Flue-gas samples were taken at six points in the uptake and collected over one-hour periods. Flue-gas temperatures were measured with thermocouples at the same six points where samples were

TABLE 1 SUMMARY OF RESULTS OF 11 STEAMING TESTS ON AN EDGEMOOR BOILER BURNING POWDERED COAL AT ONEIDA STREET POWER STATION, MILWAUKEE, WIS.

Heating Surface:
Boiler....4680
Superheater....594
Furnace coil....48
Total heating surface....5322 sq. ft.

Draft — Natural.
Burner — Lopulco vertical.
Coal feed — Screw and air blast.

Rating of boiler....468 hp.
Volume of furnace....1600 cu. ft.
Greatest height of furnace....16.7 ft.
Greatest width of furnace....9.3 ft.
Greatest length of furnace....13.3 ft.

1	Test number	28	29	30	31	32	33	34	35	36	37	38
2	Duration, hours	22.90	23.72	18.17	23.62	23.60	22.47	25.22	23.53	23.43	23.33	16.73
Coal as Fired												
3	Per cent through 100 mesh	96.10	95.80	.....	95.40	93.20	93.10	90.80	88.60	.....	94.40	95.40
4	Moisture content, per cent	1.42	2.92	2.75	2.55	3.79	3.07	3.60	3.47	7.69	8.23	8.23
5	Volatile matter, per cent	36.62	36.66	37.45	36.58	36.57	36.29	37.17	36.27	35.82	34.42	34.70
6	Fixed carbon, per cent	48.16	46.63	46.08	48.07	48.43	49.01	46.39	48.87	45.74	46.39	44.67
7	Ash, per cent	13.80	13.79	13.72	12.80	11.21	11.63	12.84	11.39	10.75	10.96	12.40
8	Sulphur, per cent	2.66	3.64	3.49	2.92	2.40	2.66	3.43	2.99	2.10	2.21	2.90
9	Calorific value, B.t.u.	11956	11860	11875	12085	12172	12178	11889	12188	11565	11508	11245
10	Total fuel fired, lb.	40214	39862	52746	46613	43947	45469	47459	41250	43729	41640	36315
11	Coal fired per hour, lb.	1756	1681	2903	1973	1862	2024	1882	1753	1866	1785	2171
12	Coal fired per hour per cu. ft. of combustion space	1.10	1.05	1.81	1.23	1.16	1.26	1.18	1.09	1.16	1.12	1.36
Ash and Refuse												
13	Carbonaceous content in furnace slag, per cent	0	0	0	0	0	0	0	0	0	0	0
14	Carbonaceous content in 2d and 3d pass refuse, per cent	4.15	3.49	5.00	5.25	9.52	7.19	7.52	7.37	4.97	4.32	4.49
15	Carbonaceous content in uptake dust, per cent	4.95	5.24	7.35	5.13	7.70	6.45	7.81	5.57	3.91	3.50	3.28
16	Calculated total carbon in refuse, per cent of coal fired	0.50	0.54	0.62	0.36	0.87	0.61	0.67	0.30	0.24	0.22	0.26
17	Softening temperature of coal ash, deg. fahr.	.....	2050	2210	2120	2120	2110	2060	.....	.....	.....	.....
Ash Account (per cent of ash fired)												
18	From bottom of furnace	29.20	25.50	41.50	48.10	10.10	24.10	37.80	59.40	48.10	47.10	41.60
19	From 2d and 3d pass	12.10	11.60	5.80	10.40	10.30	7.00	9.10	10.50	8.30	7.70	8.70
20	From dust collector	31.50	25.00	33.20	29.20	27.50	29.00	25.50	30.10	32.80	26.00	26.30
21	Unaccounted for	28.20	37.90	19.50	12.30	52.10	39.90	27.60	0.00	10.80	19.20	23.40
Air												
22	Temperature, air at furnace, deg. fahr.	83	90	80	76	85	71	64	74	79	72	75
23	Pressure air at feeder, inches of water	5.00	5.30	7.50	5.10	5.60	5.50	6.40	5.90	5.70	5.50	6.20
24	Excess air in flue gas, per cent	30	22	18	18	19	15	20	25	19	16	20
Flue Gas												
25	Carbon dioxide, per cent by volume	14.10	14.90	15.40	15.50	15.30	15.80	15.10	14.60	15.50	15.80	15.40
26	Oxygen, per cent by volume	4.80	3.80	2.90	3.30	3.20	2.40	3.60	4.10	3.20	2.90	3.60
27	Carbon monoxide, per cent by volume	0	0	0.26	0	0	0.10	0.10	0	0	0	0
28	Pounds of dry flue gas per pound of coal	12.40	11.20	10.60	11.00	11.30	10.80	11.00	11.70	10.70	10.40	10.60
29	Temperature of gases in uptake, deg. fahr.	517	492	610	483	457	472	470	486	484	466	514

Draft												
30	At uptake, inches of water	0.12	0.10	0.27	0.09	0.06	0.08	0.05	0.09	0.10	0.05	0.15
31	Top of furnace, inches of water	0.00	0.02	0.00	0.00	0.01	0.00	0.01	0.02	0.02	0.03	0.03
Steam and Water												
32	Steam pressure, lb. absolute	184	186	196	189	188	187	185	186	183	182	185
33	Degrees superheat	60	59	80	58	53	61	67	79	74	70	102
34	Temperature of feedwater, deg. fahr.	108	99	99	101	101	100	100	103	98	100	99
35	Temperature of water to coil, deg. fahr.	No coil	No coil	No coil	54	53	52	50	48	46	46	46
36	Temperature of water from coil, deg. fahr.	No coil	No coil	No coil	129	146	145	118	151	139	130	144
Rates of Heat Absorption												
37	Per cent of builder's rating (boiler only)	106.6	103.9	167.4	111.7	102.7	112.4	103.4	104.8	101.9	94.4	113.5
38	Horsepower developed (boiler only)	498.8	486.5	779.0	523.2	480.8	526.0	484.0	490.5	477.0	442.0	531.5
39	Horsepower developed (superheater only)	15.8	15.3	32.9	16.3	13.7	15.0	15.4	20.5	18.4	16.3	27.9
40	Horsepower developed (furnace coil)	No coil	No. coil	No coil	49.7	55.0	59.7	45.9	7.5[1]	32.0	29.1	26.8
41	Total horsepower developed	514	502	812	589	549	601	545	528	527	487	586
B. t. u. Absorbed per Sq. Ft. of Heating Surface per Hour												
42	By water in boiler	3567	3482	5575	3743	3437	3758	3466	3508	3413	3207	3798
43	By steam in superheater	892	861	1856	920	772	845	869	1155	1039	934	1573
44	By water in furnace coil	No coil	No coil	No coil	33900	37600	40770	31270	33190	21920	20160	18440
Heat Absorbed per Pound of Coal as Fired, B. t. u.												
45	By water in boiler	9500	9690	8990	8870	8630	8710	8625	9360	8560	8405	8190
46	By steam in superheater	301	304	380	277	246	249	274	391	330	311	430
47	By water in coil	No coil	No coil	No coil	843	995	988	815	335	575	554	414
48	Total absorbed	9801	9994	9370	9990	9871	9947	9714	10086	9465	9270	9034
	HEAT BALANCE (Per cent of heat in coal fired)											
Heat Absorbed												
49	By water in boiler	79.4	81.6	75.6	73.4	70.8	71.5	72.4	76.8	74.0	73.0	72.7
50	By steam in superheater	2.5	2.5	3.2	2.3	2.0	2.0	2.3	3.2	2.8	2.7	3.8
51	By water in coil	No coil	No coil	No coil	7.0	8.2	8.1	6.9	2.7	5.0	4.8	3.7
52	Total thermal efficiency	81.9	84.1	78.8	82.7	81.0	81.6	81.6	82.7	81.8	80.5	80.2
Heat Carried Away												
53	By dry gases	10.8	9.1	11.4	8.9	8.2	8.6	9.0	9.5	9.0	8.5	9.9
54	By steam from burning hydrogen	4.1	4.3	4.2	4.1	4.1	4.2	4.2	4.0	4.2	4.1	4.2
55	By steam from moisture in coal	0.1	0.3	0.3	0.2	0.3	0.3	0.3	0.3	0.8	0.9	0.9
56	By steam entering with air	0.3	0.3	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1	0.1
57	By carbon monoxide	0.0	0.0	1.0	0.1	0.1	0.2	0.4	0.0	0.0	0.0	0.0
58	By carbon in ash and flue dust	0.6	0.6	0.7	0.4	1.0	0.7	0.8	0.4	0.3	0.3	0.3
59	By radiation	2.5	2.6	1.9	2.2	2.3	2.1	2.3	2.4	2.4	2.5	2.2
60	Heat unaccounted for	— .3	—1.3	1.6	1.3	2.9	2.2	1.3	0.6	1.4	3.1	2.2
61	Total	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0

[1] Cooling coil in operation during first 8½ hours of test only.

drawn for analysis, and readings were taken every 15 min. The flue-gas temperature given Table 1 is the average of the measurements with the six couples.

RESULTS OF THE TESTS

8 The results of the tests are given in Table 1. The quantities of heat absorbed by the boiler, superheater and cooling coil, when the latter was used, are itemized separately. In the heat balance the losses by radiation are given by a separate item. In a series of tests on the same boiler and setting the radiation loss per square foot of exposed surface should be nearly constant and should vary only slightly by the capacity developed by the boiler. For the calculation of the radiation loss it was estimated that 250 B.t.u. were lost per sq. ft. of the exposed surface per hour when the boiler was operated at 100 per cent of rating, and 350 B.t.u. when operated at 200 per cent of rating. The radiation loss was calculated according to the percentage of rating developed. These calculations of the radiation loss leave the true "unaccounted for," which consists largely of errors. In a series of well-conducted boiler tests this true "unaccounted for" should be close to zero and should vary on both sides of the zero line according to whether the plus or minus errors predominate.

EFFECT OF FINENESS ON RESULTS OF TESTS

9 It has been customary to state that in order to get good results the coal must be pulverized to a fineness of 95 per cent through a 100-mesh screen and 85 per cent through a 200-mesh screen. Table 2 gives the results of complete sizing tests of the coal burned in Tests Nos. 32 to 35, inclusive. The coal was much coarser than specified by the foregoing statement. The results of these tests seem to indicate that it is not necessary to pulverize the coal to the extreme fineness of 85 per cent through a 200-mesh screen in order to get good combustion and good efficiency. The completeness of combustion seems to be more a matter of a proper furnace and burner design and the right way of supplying air, than of the fineness of the coal. The losses due to coarseness of coal would be shown by the greater percentage of carbon in the refuse. The average loss due to this cause for the four tests with the coarser coal is 0.7 per cent. The averages of this loss for the previous four tests is 0.6 per cent. The averages of the efficencies are very nearly the same.

10 The ability to burn coarser coal means increased capacity of the pulverizing mills and decreased cost of coal preparation.

EFFECT OF MOISTURE IN COAL ON RESULTS OF TESTS

11 Another statement that has been generally accepted is that coal must be dried to about 1 per cent moisture in order to be successfully burn in pulverized form. In order to determine to what extent this statement is true, tests Nos. 36, 37 and 38 were run with undried coal. The results of the tests show that the completeness of combustion was as good as with the dried coal. There was no loss

TABLE 2 RESULTS OF SIZING TESTS OF COAL BURNED IN TESTS NOS. 32-35

Test No.	Percentage of Coal Passing Through Screens			
	20-mesh	40-mesh	100-mesh	200-mesh
32	99.9	99.2	93.2	67.0
33	99.9	99.2	93.1	70.1
34	100.0	98.9	90.8	65.5
35	99.8	98.0	88.6	64.0

due to CO in the flue gases and the losses due to combustible in the refuse averaged only 0.3 per cent for the three tests, which is in fact less than the average with the dried coal.

12 The losses due to moisture in coal of course increased 0.5 to 0.6 per cent, which increase is at the rate of about 0.1 per cent for every 1 per cent of increase of moisture in the coal. The average decrease in the boiler efficiency for the three tests is about 0.7 per cent, which checks closely with the increase in the losses due to increased moisture in the coal. It seems, therefore, that it is not necessary to dry the coal down to 1 per cent of moisture in order to get good boiler efficiency. In fact, it seems that most of the eastern coals can be pulverized and burned with good results without drying.

CAPACITY OF BOILER THAT CAN BE DEVELOPED WITH PULVERIZED COAL

13 The capacity of boiler that can be developed with pulverized coal depends entirely upon the size and shape of the furnace.

With the present knowledge of the art of burning powdered coal the best results are obtained when the coal is burned at the rate of 1 to 1½ lb. per cu. ft. of combustion space per hour. Good results can be obtained when the coal is burned at rates varying from ½ to 2lb. per cu. ft. of combustion space per hour, which gives a considerable working range. In Table 1 the rate of combustion is given by item 12. The range covered by this series of tests is from 1.05 to 1.81 lb. of coal per cu. ft. of combustion space. If it is desired to operate the boiler at high rates of working, a large furnace must be installed and the combustion space must be so arranged that the flames are given the longest possible path through the furnace. The design of burners and the admission of air are very important at high rates of combustion. It appears probable that future developments in the design of furnaces, burners and the air supply may make possible higher rates of combustion than the limit given above.

DISCUSSION

GUILDFORD GREIG and W. C HECKEROTH. For about two years the engineering department of the Erie City Iron Works has experimented with and successfully burned pulverized coal under a 402-hp. Erie City horizontal water-tube boiler at their works.

In the face of much criticism, we have persisted in the practice of pulverizing coal of any moisture content, thus eliminating drying equipment. In our experimnets we have also found it quite practical to burn the pulverized coal efficiently without reducing it to the extreme fineness heretofore considered necessary, and it is certainly gratifying to find definite confirmation of the practicability of our methods by such eminent authorities as the authors of the present paper.

The extreme simplicity of our apparatus enabled us to obtain data with a minimum of effort and no mechanical complications, other than those required for an ordinary boiler test complying with the A.S.M.E. test code.

As can be seen in Fig. 3, the only apparatus required by our method of preparing coal for burning in pulverized form other than that required in ordinary stoker-fired boiler plants is a magnetic separator and a pulverizing machine. The magnetic separator removes tramp iron from the coal before it reaches the pulverizer.

The pulverizer is a very simple machine, consisting of a cylin-

drical housing, containing a rotor mounted on a shaft, the rotor being composed of pulverizing elements and a fan.

The coal, of a size that will pass through a 2-in. ring, and of any moisture content, is fed to the pulverizer through a simple adjustable feeding device which is integral with it. The pulverizer reduces the coal to a powder, approximately 95 per cent through a 200-mesh screen. These degrees of fineness, however, are not essential to efficient operation.

The fan serves two purposes; first, inducing the flow of a

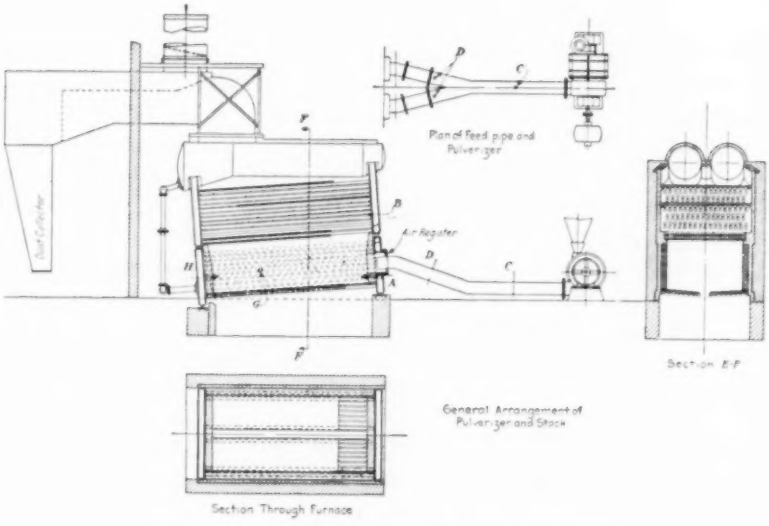


FIG. 3 PULVERIZED FUEL BURNING FURNACE

regulable volume of air with the coal, sufficient to facilitate the travel of coal through the pulverizer, and second, to supplement this volume of air with enough additional air to support combustion. This supplemental air is admitted to the fan through an orifice in the fan housing where air and pulverized coal are thoroughly mixed before being delivered to the furnace. Definite regulation of air is easily accomplished by manipulation of the adjustable damper on the fan housing. All air entering the furnace enters in mixture with the pulverized coal and this mixture is delivered to the furnace from the pulverizer through a plain open-end pipe.

The furnace is of the Seymour type with water-cooled refactories which make it possible to burn pulverized coal with only

five to ten per cent excess air, thus reducing stack losses to a minimum.

Referring to Fig. 3, it will be seen that the water cooling of the refactory lining is accomplished by the circulation of water through tubes and headers disposed in side walls, end walls and pavement of furnace, which become an integral part of the boiler, being connected thereto with pipes and nipples. The pavement of the furnace inclines from each side toward the center, thus compelling the movement of the molten slag and ash to the ashpit through an opening in the furnace pavement. The ashpit is cooled by the

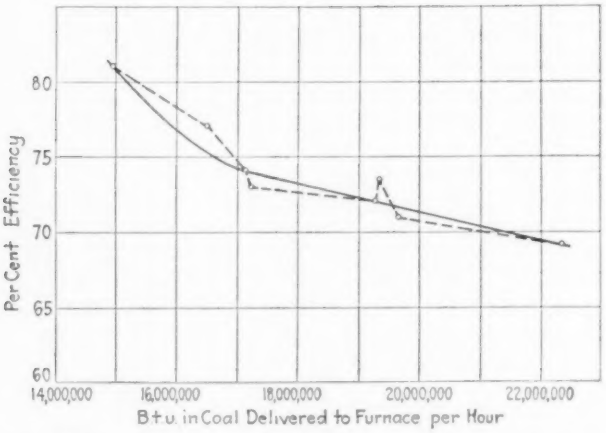


FIG. 4 EFFICIENCY OF BOILER AND FURNACE, PULVERIZED FUEL

water tubes which form the pavement of the furnace, thus making the ash friable and easily removable. The water cooling of the ashpit is a patented feature of the system.

It has been found that the best results are obtained in this type of furnace with a volume of $1\frac{3}{4}$ cu. ft. per horsepower developed.

During tests no trouble has been experienced because of slagging of the boiler tubes.

The total power consumption for preparing the coal and delivering coal and air to the furnace, during the tests, expressed as a percentage of the fuel burned equals approximately 1.9 per cent.

H. G. BARNHURST. The tests reported in the paper show economies which the writer has always claimed possible by the use of pulverized fuel. The authors make certain statements which

must be subjected to careful consideration before they can be accepted as final and conclusive as to the methods which should be used in the preparation of coal in pulverized form.

The authors mention the fact that a general recommendation has been made that coal should be pulverized to 95 per cent through a 100-mesh and 85 per cent through a 200-mesh screen. The writer believes that these recommendations, particularly during the present stage of development of this method of burning, should be followed, so as to cover all coals. Just how any particular coal should be prepared depends upon the coal in question, particularly as to the amount of volatile and ash in the coal.

In a number of tests of a plant using pulverized anthracite fuel, published by the M. A. Hanna Co. of Cleveland and made on one of their boilers at the Lytle Coal Co.'s plant at Lytle, Pa., a distinct increase in efficiency was obtained as the fineness of the coal increased. The coal used in these tests averaged only 8.65 per cent volatile, 78 per cent fixed carbon, and between 11 and 12 per cent ash. The tests quoted in the present paper show that the coal has been pulverized to a fairly high degree, and therefore their statements regarding the burning of coarser coals other than those generally specified do not mean that this coal is at all of a coarse character, and it does not mean that the coal can be granulated for burning in a pulverized form as is now being recommended by some engineers.

It is impossible to obtain satisfactory results unless there is a large percentage of 100- and 200-mesh material in the pulverized coal. There cannot be any doubt but that the finer the coal is pulverized the greater is the surface exposure, the better is the mixture of the particles with the air for combustion, the quicker is the ignition, the lower will be the percentage of combustible in the ash, and the higher will be the efficiency obtained, provided, of course, correct mxing and burning equipment is installed. Therefore, it appears to the writer, that recommendation should be made at the present time that machinery installed should have the ability to pulverize the coal to a fineness of 95 per cent through the 100-mesh screen and 85 per cent through the 200-mesh screen, and then, afterwards, each individual installation can determine to what degree of coarseness its particular coal can be pulverized, everything affecting the economy of operation being taken into consideration.

In reference to moisture, mention should also be made of the surface and the inherent moisture so these will not be confused, for

TABLE 3 DATA AND RESULTS OF EVAPORATIVE TESTS ON A 402-HP ERIE CITY WATER-TUBE BOILER USING PULVERIZED BITUMINOUS COAL

		Test No.							
		1	2	3	4	5	6	7	8
1	Test of No. 11–402-hp. horizontal water-tube boiler located at East Power House, E. C. I. Works, to determine efficiency, conducted by W. C. Heckeroth								
2	Kind of furnace, pulverized coal								
3	Grate surface, sq. ft., none	4022	4022	4022	4022	4022	4022	4022	4022
4	Water heating surface, sq. ft.								
5	Superheating surface, sq. ft., none								
6	Date	4-8-21	4-12-21	4-13-21	4-14-21	4-15-21	4-20-21	4-21-21	4-22-21
7	Duration, hr.	8	8	8	8	8	8	8	8
8	Kind and size of coal, Pittsburgh bituminous slack								
9	Steam pressure by gage, lb.	146.5	149.3	151.1	151.3	148.6	145.8	146.2	149.4
10	Temperature of feedwater entering boiler, deg. fahr.	51	50	50	50	51	49	51	51
11	Temperature of escaping gas leaving boiler, deg. fahr.	473	451	469	479	466	480	488	415
12	Force of draft between damper and boiler, in. at outlet	0.408	0.544	0.396	0.13	0.209	0.198	0.212	0.197
	Force of draft between damper and boiler, in. in furnace	0.088	0.097	0.088	0.038	0.06	0.042	0.05	0.036
13	Percentage of moisture in steam	2.11	2.01	1.81	1.85	1.76	1.78	1.738	1.58
14	Weight of coal as fired,[1] lb.	14119	15306	14570	12050	12470	12400	12850	13185
15	Percentage of moisture in coal	7.4	6.5	9.0	8.9	8.45	7.75	8.43	7.9
16	Total weight of dry coal consumed, lb.	13074	14311	13259	10978	11416	11439	11767	12143
17	Total ash from chemist's analysis, lb.	2402	2095	2352	2443	2196	1932	2255	1574
18	Percentage of ash in dry coal from chemist's analysis, per cent	18.37	14.64	17.73	22.25	19.23	16.89	19.16	12.96
19	Total weight of water fed to the boiler,[2] lb.	98715	107240	96442	84284	88213	88021	87192	95809
20	Total water evaporated, corrected for moisture in steam, lb.	96632	105083	94696	82725	86660	86454	85677	94295
21	Total equivalent evaporation from and at 212 deg., lb.	117057	127434	114857	100337	104997	104895	103787	114155
22	Dry coal consumed per hr., lb.	1634.2	1788.9	1657.4	1370.7	1427	1430	1471	1518
23	Dry coal per sq. ft. of grate surface per hr., lb.								
24	Water evaporated per hr. corrected for quality of steam, lb.	12079	13135	11837	10341	10832	10807	10709	11787
25	Equivalent evaporation per hr. from and at 212 deg., lb.	14632	15929	14357	12542	13125	13112	12973	14269
26	Equivalent evaporation per hr. from and at 212 deg. per sq. ft of water heating surface, lb.	3.513	3.96	3.569	3.118	3.263	3.26	3.22	3.52
27	Evaporation per hr. from and at 212 deg., lb.	14632	15929	14357	12542	13125	13112	12973	14269
28	Boiler horsepower developed	424.1	461.7	461.1	363.5	380.4	380.0	376.0	413.5
29	Rated capacity, in evaporation from and at 212 deg. per hr., lb.	13876	13876	13876	13876	13876	13876	13876	13876
30	Rated boiler horsepower	402.2	402.2	402.2	402.2	402.2	402.2	402.2	402.2
31	Percentage of rated capacity developed	105.4	114.7	103.4	90.38	94.58	94.48	93.48	102.7
32	Water fed per lb. of coal fired (Item 19 ÷ Item 14), lb.	6.991	7.0	6.62	6.994	7.074	7.098	6.786	7.266
33	Water evaporated per lb. of dry coal, lb.	7.391	7.342	7.14	7.54	7.591	7.558	7.281	7.767
34	Equivalent evaporation from and at 212 deg. per lb. of dry coal	8.956	8.9	8.662	9.148	9.197	9.17	8.82	9.4
35	Equivalent evaporation from and at 212 deg. per lb. of combustible	10.968	10.431	10.53	11.75	11.39	11.033	10.911	10.80
36	Calorific value of dry coal, B.t.u. per lb.	11823	12501	11849	10938	11577	11993	11711	12677
37	Calorific value of combustible, B.t.u. per lb.	14483	14645	14403	14068	14334	14430	14487	14565
38	Efficiency of boiler, furnace and grate	73.49	69.12	70.94	81.05	77.1	74.19	73.08	71.95
39	Efficiency of boiler and furnace	1.21137	1.2127	1.2129	1.2129	1.2116	1.2133	1.21137	1.21168
40	Factor of evaporation								

[1] The term "as fired" means actual condition including moisture, corrected for estimated difference in weight of coal on the grate at beginning and end.
[2] Corrected for inequality of water level and steam pressure at beginning and end.

TABLE 4 SPECIAL DATA ON FURNACE AND PULVERIZER USED IN CONNECTION WITH TESTS OF TABLE 3

Test No.	1	2	3	4	5	6	7	8
Date	4–8–21	4–12–21	4–13–21	4–14–21	4–15–21	4–20–21	4–21–21	4–22–21
Highest furnace temperature at "G" deg. fahr.	2155	2560	2547	2625	2690	2720	2805	2750
Lowest furnace temperature at "G" deg. fahr.	1737	2315	2315	2404	2305	2480	2495	2625
Average furnace temperature at "G" deg. fahr.	1980	2465	2429	2514	2586	2625	2677	2683
Average stack temperature, deg. fahr.	473	451	469	479	466	480	488	515
Average speed of pulverizer, r.p.m.	946.3	954.1	949.7	961.6	952.8	953.6	939	945.4
Volume of furnace, cu. ft.	735	735	735	735	735	735	735	735
Volume of furnace, cu. ft. per rated hp.	1.827	1.827	1.827	1.827	1.827	1.827	1.827	1.827
Volume of furnace, cu. ft. per hp. developed	1.732	1.592	1.766	2.022	1.932	1.934	1.954	1.777
Coal as fired, lb. per hr.	1765	1913.2	1821.2	1506.2	1558.8	1550	1606.2	1648.1
Coal as fired, lb. per boiler hp. per hr.	4.158	4.143	4.376	4.144	4.097	4.079	4.271	3.985
Coal, dry basis, lb. consumed per hr.	1634.2	1788.9	1657.4	1370.7	1427	1430	1471	1518
Coal, dry basis, lb. consumed per boiler hp. per hr.	3.85	3.874	3.983	3.77	3.751	3.763	3.912	3.671
Coal as fired, lb. per kw. consumed	100.57	108.76	106.87	98.0	98.34	90.85	83.57	85.75
Coal, dry basis, lb. per kw. consumed	93.11	101.7	97.06	89.18	90.0	83.82	76.53	76.85
Kw-hr. consumed per ton of 2000 lb. of wet coal pulverized	19.88	18.38	18.71	20.4	20.33	22.01	23.93	23.34
Hp-hr. consumed per ton of 2000 lb. of wet coal pulverized	26.64	24.63	25.07	27.34	27.25	29.49	32.06	31.27
Kw-hr. consumed per ton of 2000 lb. of coal pulverized, dry basis	21.47	19.66	20.56	22.42	22.21	23.86	26.13	26.02
Hp-hr. consumed per ton of 2000 lb. of coal pulverized, dry basis	28.77	26.52	27.54	30.04	29.77	31.97	35.01	34.86
Analysis of coal as received								
Moisture, per cent	7.4	6.5	9.0	8.9	8.45	7.75	8.43	7.9
Volatile, per cent	29.64	31.67	29.56	28.08	29.72	30.5	28.98	32.36
Fixed carbon, per cent	45.95	48.14	45.3	42.75	44.22	46.17	45.04	47.8
Ash, per cent	17.01	13.69	16.14	20.27	17.61	15.58	17.55	11.94
Calorific value, B.t.u. per lb. dry coal	11823	12501	11849	10938	11577	11993	11711	12677
Gas analysis taken at B, CO_2	13.3	14.4	15.4	16.9	16.7	15.6	16.1	16.05
O_2	4.37	2.19	2.19	0.79	0.91	1.79	1.58	1.7
CO	0.05	0.047	0.053	0.052	0.047	0.028	0.048	0.027
Excess air, per cent	25.36	11.11	11.27	3.79	4.38	9.0	7.89	8.547
Velocity of air through discharge pipe at "C," ft. per min.	3806.7	3657.2	3486.5	2689.1	2800	2907.3	2982	3071.3
Velocity of air through discharge pipe at "D," ft. per min.	3348.6	3217	3066.9	4730.4	4925.6	5114.8	5246	5415.8
Draft								
At damper	0.408	0.544	0.396	0.13	0.209	0.198	0.212	0.197
In furnace	0.088	0.097	0.088	0.038	0.06	0.042	0.05	0.036
Loss between damper and furnace	0.320	0.447	0.308	0.092	0.149	0.156	0.162	0.161

Tests beginning with April 14 were run with one branch only of pipe "*D*" in operation. (See Fig. 3.)

Screen tests not given because powdered coal samples were unreliable due to difficulty in getting representative samples at low velocity in discharge pipe.

the reason that while the writer advocates that the surface moisture be reduced to approximately 3 per cent, generally speaking, it is not necessary to eliminate the inherent moisture.

All pulverized coal plants should be equipped with driers to dry the coal before preparation. The coal should be dried not only on account of the burning conditions in the furnace, but primarily to eliminate the effect that moisture has on the general handling of coal in transportation, storage and feeding to furnaces. Driers should also be installed to take care of the variation in the moisture content that occurs throughout the year. They should be installed particularly to take care of exceptional conditions which almost always occur and when least expected.

The writer also notes with interest that the results of these tests prove that the successful use of pulverized coal in boiler furnaces depends strictly upon the furnace proportions. He has always recommended large furnaces, proportioned on a basis of 2 cu. ft. per boiler horsepower developed for the horizontal method of firing, and 3 cu. ft. for the vertical method, basing recommendations on the erosive effect of the gases of combustion passing through the furnace. These recommendations, however, are only general in their nature, for if the successful use of pulverized fuel is dependent upon having the furnaces so proportioned that the velocity of the gases will not create erosion and destruction, then it will be necessary to design the furnaces for the rating required based on the amount of combustible that will be burned. The writer notes that the recommendations of the authors come within the range of his recommendations.

The writer is heartily in accord with the statements made regarding burners and the admission of air. These are two of the most important features that must be considered in connection with the burning of pulverized coal. These tests show that higher efficiencies can be obtained by this method, and the writer believes that ultimately coal in pulverized form will be used extensively.

FRED'K A. SCHEFFLER. This brief paper is one of the most important articles that has been presented in recent years on the interesting subject of firing boilers with pulverized coal, and is all the more of interest because of the thoroughness with which the tests were carried out in every detail and the records tabulated.

The writer is particularly glad to have confirmation by these tests of the tabulation which he made for the Fuller Engineering

Company for their boiler bulletin No. 600, of eighteen other pulverized-coal-fired boiler tests by various parties through the country and with various kinds of fuel and boilers, which showed an average of 80.5 per cent boiler and furnace efficiency.

The present paper shows an average of eleven tests, exclusive of the water coil in the ashpit, of 77.2 per cent boiler and furnace efficiency, so that there are now twenty-nine recorded tests of pulverized coal under boilers using Renton buckwheat from the vicinity of Seattle, Kansas bituminous, Gallup, New Mexico, Issaquah (vicinity of Seattle) screenings, Nanaimo slack from Vancouver Island, B. C., Pennsylvania bituminous, Illinois coals, etc., all of which show an average of 78.8 per cent boiler and furnace efficiency.

From the above it is evident that 80 per cent efficiency can be safely considered the boiler and furnace efficiency with any kind of coal, properly prepared in pulverized form, when burned under boilers that are properly baffled and with proper design of furnace to suit the operating loads and conditions.

G. E. Pfisterer. The outstanding feature in the writer's opinion is the fact that the paper demonstrates the fallacy of the idea that the coal must be pulverized to a fineness heretofore presumed in order to obtain good results. It may be hoped that future tests, with probably different burners and different furnace designs may still further reduce the fineness of coal. Probably it also may be demonstrated later that undried coal can be burned to advantage. These are all steps in the proper direction, for if the cost of drying and pulverizing can be reduced, or eliminated, the field of usefulness of pulverized coal can be extended.

It would be interesting to know what the maintenance of the furnace amounts to and what effect on maintenance the introduction of the water coil had. Also, what was done with the heated water after leaving the pipe coils. Since the heat absorption of the coil is very large per square foot, might it not be possible to put this coil in circulation with the boiler? From Test No. 30 it appears that when the combustion rates are increased to give a boiler rating of approximately 167 per cent, the exit flue gas temperature rises notably, and the efficiency falls off. The presence of CO is also noted, which seems to indicate incomplete combustion. For widely varying loads, therefore, it appears that the combustion space must be amply large, for with forced-draft stokers the operating range is quite large.

The unaccounted ash evidently goes up the stack with the the flue gas as finely divided or powdered ash. With a dust collector a great deal of the dust can be collected, but if the plant were located in a densely populated district with surrounding buildings, this dust might be objectionable.

When burning pulverized coal with its small excess of air, it may become necessary to arrange the boiler baffles so as to give proper velocity of gas through the tubes and increase heat absorption. The draft loss through the boiler was undoubtedly very low, indicating low velocity of gas. If the gas velocity were increased, might we expect better absorption per square foot of heating surface?

The writer believes that it is necessary to have more experimental data before this method of firing can be put on the same basis as stoker firing.

JOHN A. HOFF. It would appear from the summary of the tests that a 468-hp. Edgemoor boiler developed a thermal efficiency of 80.5 per cent. However, the facts and figures also stated show that the tests were actually made of two boilers in one setting, the Edgemoor boiler developing 75.7 per cent efficiency at 113 per cent rating, and a secondary boiler or coil of 4.8 hp. developing a thermal efficiency of 4.8 per cent at 550 per cent of rating.

The water feed of the secondary boiler entered at 46 deg. fahr. from another source, was raised to 144 deg. and delivered to an unstated destination. The boiler feed-water entering the Edgemoor boiler was taken in at 99 deg. fahr. instead of 144 deg., the coil discharge, involving a heat loss or waste of 44 B.t.u. per lb. of water evaporated by the boiler. If any data accounting for the above discrepancies are available, they would be of interest.

VICTOR J. AZBE. The writer has long held that the ideal efficiency is 89.93 per cent, obtainable with 15 per cent excess air and 250 deg. flue gas temperature. The authors have succeeded in obtaining 90 per cent overall boiler efficiency.[1] If such efficiency is possible, we should cease operating boilers at 50 and 60 per cent efficiency as is usually done.

C. C. TRUMP. It seems to the writer that there is room for improvement in the pulverizing, handling and drying equipment for pulverized fuel and that the first cost of the installation might be reduced. A great advantage of the pulverized-fuel furnace is

[1] See authors' closure for report of these tests.

the radiant heat from the flame as the fuel is burned in suspension, and this results in greater efficiency of heat absorption by the boiler.

E. N. TRUMP. Ninety per cent efficiency is not unattainable by any means. The writer has obtained such an efficiency with large size boilers and underfeed stokers, the average of a month's run. This includes economizers as well as boilers and is with about 12 per cent CO_2. This efficiency is high, but with care might be exceeded; 86 and 87 per cent are quite usual.

R. SANFORD RILEY. The authors have set a new standard for combustion, one which is a little higher than what we have had before. Before long we shall see some developments in stokers which will substantiate what Mr. Trump has said. The limitations of powdered fuel would form a basis for profitable discussion, and not the least of these limitations is the necessary investment.

H. KREISINGER. Mr Hoff brought up some points with reference to the furnace coil which need explanation. The furnace coil was an experimental apparatus devised to keep the ash which falls to the bottom of the furnace from fusing. Water was supplied to this coil separately in order to make it possible to determine the rate of heat absorption. This information was desirable for the determination of the proper size of the coil for keeping the bottom of the furnace below the fusion point of the ash. The water from the coil was discharged into the hot well from which it was pumped into other boilers in the plant. The test boiler was supplied with water at a lower temperature than that of the water in the hot well in order to make its weighing more convenient. Later the boilers at this plant were equipped with coils directly connected in the circulation, so that the coils are part of the boilers.

Mr. Hoff refers to the percentages of heat absorbed by the different pieces of the apparatus as thermal efficiencies. Such reference is rather misleading; the term "thermal efficiency" should be reserved for total percentage of heat absorbed by the combined apparatus. It would be difficult to arrive at a logical value that would express the true thermal efficiency of the different pieces of apparatus. Because the boiler absorbed 73.0 per cent, and the superheater 2.7 per cent of the heat in the coal burned, it would be hardly right to say that the thermal efficiency of the boiler was 73.0 per cent and that of the superheater was 2.7 per cent. Similarly it is hardly proper to speak of the thermal efficiency of the

furnace coil as being 4.8 per cent. The thermal efficiency of any piece of apparatus might be defined as the amount of heat absorbed by it divided by the amount of heat available for it for absorption. It is difficult to determine this amount of heat that was available for each piece of the apparatus. Surely not all of the heat in the coal was available for each piece of the apparatus. For this reason in the table of results the precentages of heat absorbed by the different pieces of apparatus are given under the caption of "Heat Absorbed." Only the total heat absorbed is termed the thermal efficiency. Tests 28, 29, and 30 show what results were obtained when there was no coil in the furnace.

Table 5 presents additional data on pulverized fuel plants and the efficiencies obtainable.

TABLE 5 SUMMARY OF RESULTS OF FIVE BOILER TESTS WITH PULVERIZED ILLINOIS COAL. TESTS MADE ON BOILER NO. 8, LAKESIDE STATION OF THE MILWAUKEE ELECTRIC RAILWAY AND LIGHT CO.

Test No.	1	2	3	4	5
Duration, hr.	42.33	23.97	19.92	24.20	24.17
Coal as Fired:					
Per cent through 100 mesh	89.2	90.8	90.5	92.2	90.5
Per cent through 200 mesh	67.7	68.7	69.1	70.5	66.7
Moisture content, per cent	2.25	3.56	3.59	5.24	5.61
Volatile matter, per cent	36.60	36.48	35.66	36.30	35.85
Fixed carbon, per cent	49.60	48.33	48.70	46.10	47.16
Ash, per cent	11.55	11.63	12.05	12.36	11.38
Sulphur, per cent	2.26	2.74	2.28	3.91	3.39
Hydrogen, per cent	4.97	5.18	5.01	5.10	5.06
Carbon, per cent	68.88	67.20	66.22	63.44	65.41
Calorific value, B.t.u.	12321	12022	11917	11483	11661
Total fuel fired, lb.	253161	233477	190334	160881	274640
Fuel fired hourly, lb.	5980	9740	9560	6650	11350
Fuel fired hourly per cu. ft. combustion space, lb.	0.85	1.39	1.36	0.95	1.62
Ash and Refuse:					
Carbon in second- and third-pass refuse, per cent of coal fired	3.60	5.06	5.06	5.31	7.17
Carbon in uptake dust, per cent	6.26	5.09	3.39	2.82	5.95
Unburned carbon per lb. coal, per cent	0.45	0.45	0.40	0.40	0.74
Ash Account:					
From bottom of furnace, lb.	6240	6414	5230	4420	1000
From second and third pass, lb.	13565	12236	9278	8260	12270
Determined from dust-collector data, lb.	10580	8640	8632	7290	18120
Air:					
Temp. of air entering furnace, deg. fahr.	86	89	91	104	93
Pressure of air at feeders, in. of water	11.0	12.2	13.6	12.2	13.6
Air entering with coal, lb. per lb. coal	2.05	1.25	1.27	1.85	1.03
Air entering at burners, lb. per lb. coal	.....	.....	.....	2.07	.....
Air through hollow wall, lb. per lb. coal	.....	.....	.....	5.64	.....
Excess air in flue gases, per cent	13.3	21.9	21.2	10.7	25.2

No. 1787

LIMITATIONS OF MECHANICAL STOKERS UTILIZING MID-WEST COALS

By Edward H. Tenney, St. Louis, Mo.
Member of the Society

In this paper the author discusses the limitations imposed on mechanical stokers by the use of Mid-West coals, which are characterized by higher moisture, volatile and ash contents as compared with Eastern Coals; the limitations imposed by air supply and by the design of the furnace; and the effect of these various limitations on stoker operation.

It is the author's belief that in general the application of mechanical-stoking equipment to the use of Mid-West coals has proved to be eminently successful. Specially designed arrangements for air supply and to facilitate ignition of the fuel have been provided; and furnace volumes and gas passes have been scientifically studied, with the result that the most difficult problems — namely, those incident to operation at high rates of combustion — have been satisfactorily solved.

A CONSIDERATION of the limitations of mechanical stokers imposed by the use of Mid-West coals is pertinent at this time on account of the increasing application of mechanical-stoker equipment throughout the Middle West, and on account of the tendency on the part of some to overlook the fact that such limitations exist.

2 The problem of adjusting stoker operation to any given coal involves three fundamental considerations:

Fuel
Air Supply
Design of Furnace.

All important limitations to which the operation may be subjected will therefore be found under these heads.

LIMITATIONS IMPOSED BY THE FUEL

3 Mid-West coals are classed as bituminous or sub-bituminous, and are characterized by the higher percentages of moisture, volatile, and ash which they contain as compared with Eastern coals:

Presented at the Spring Meeting, Chicago, Ill., May 23 to 26, 1921, of The American Society of Mechanical Engineers, New York.

Constituents of Coal	An Eastern Coal[1]	A Mid-West Coal[2]
Moisture, per cent	2.21	11.90
Volatile, per cent	15.78	31.06
Fixed carbon, per cent	71.65	41.62
Ash, per cent	10.36	15.42
	100.00	100.00

[1] Pocahontas. [2] Southern Illinois.

4 On account of these characteristics they are inherently difficult of satisfactory handling on a stoker, but they present possibilities for even greater operating economies than in the case of Eastern coals because of the opportunities provided for even firing, close regulation of air supply, high temperatures, and a continuous and regular elimination of ash.

5 *Moisture.* The moisture content in Mid-West coals ranges from 6 per cent to 16 per cent, depending largely upon the amount of surface moisture which may be present. This moisture absorbs heat from the furnace for its vaporization and superheating and ends to reduce furnace temperatures, often to such an extent as to be the limiting factor in the obtaining of proper ignition at high rates of combustion.

6 *Volatile.* High volatile content is the one redeeming feature of the Mid-West coals, inasmuch as this quality tends to facilitate ignition. With other furnace conditions favorable to the proper combustion of the liberated gases, there is little doubt but that the limit of stoker capacity is increased by the increased percentages of volatile.

7 *Ash.* The ash content of these coals varies from 10 per cent to 16 per cent, depending upon the method of mining and storing, and presents one of the most difficult limitations to their use on mechanical stokers. Large quantities of incombustible matter interfere with combustion if allowed to remain in the active fuel bed. This is increasingly true at high ratings, especially so on account of the low fusing temperature of the ash, which results in the formation of clinker and the prevention of proper air supply. The efficiency of the combustion process and the effectiveness of the stoker for burning large quantities of coal at high rates of combustion are seriously limited by this clinkering quality in the Mid-West coals. This characteristic can be closely approximated from

an analysis of the ash, although the grouping of elements and the reactions at different temperatures vary to such an extent that no rule can be applied. Table 1, prepared by Mr. Edwin Lundgren[1], shows the increase toward fusibility as percentages of certain constituents change, more particularly the ferric oxide. Ash from Mid-West coals comes between "clinkering" and "bad" in the table.

TABLE 1 TYPICAL PERCENTAGE ASH ANALYSES

Composition	Non-Clinkering	Fair	Clinkering	Bad
Silica (SiO_2)	54.67	46.23	46.40	43.50
Aluminum (Al_2O_3)	41.95	31.93	16.45	17.10
Ferric Oxide (Fe_2O_3)	Trace	14.54	18.15	28.10
Calcium Oxide (CaO)	1.82	5.04	11.80	5.30
Magnesia (MgO)	1.42	2.26	4.63	0.75
Sulphur	0.55	1.50	3.00	2.70

LIMITATIONS IMPOSED BY AIR SUPPLY

8 The subject of air supply to stokers utilizing Mid-West coals is of equal fundamental importance to that of the fuel itself. Sufficient air, of course, must be supplied to furnish oxygen for the complete combustion of the carbon, hydrogen, and sulphur of the coal. In the burning of low-grade coals the distribution of air through a fuel bed heavy with ash and clinker is not conducive to its proper mixture with the combustible gases, with the result that combustion is incomplete when the gases have passed from the furnace. Such coals require large quantities of excess air, which, on account of the interference set up by the fuel bed, tend to travel in stratified streams The mixing, therefore, is incomplete and the combustion slow. Where natural draft is utilized, it is obvious that air supply is one of the important limiting factors as to efficiency in the use of these coals. Where artificial draft is utilized, air supply is seldom a limiting factor. Care must be exercised, however, in the specification of equipment for varying qualities of fuel in order that provision may be made for sufficient capacities when burning low-grade fuels at high ratings. This involves the use of quantities of excess air at high temperatures and may become a serious limiting factor.

[1]Vice-President and Chief Engineer, Frederick Engineering Co., Frederick, Md.; Assoc-Mem. Am.Soc.M.E.

LIMITATIONS IMPOSED BY THE DESIGN OF THE FURNACE

9 The furnace itself presents another important limiting factor in the utilization of low-grade fuels. The size of the combustion chamber, in that it determines the length of time during which the mixture of combustible gases and air will remain in the furnace, determines the amount of fuel that can be burned, and hence determines stoker capacity. For high ratings where large quantities of gas are produced and large volumes of air supplied, a limit is reached when the furnace gas and the oxygen in the air supply cannot remain in contact a sufficient time to completely burn the gases. With 50 per cent excess air and burning at the rate of 50 lb. per sq. ft. of grate area per hour, developing to within one-half of one per cent complete combustion of the flue gases, the Bureau of Mines[1] have determined that Illinois coal requires 11.9 cu. ft. of combustion space per sq. ft. of grate area as compared with 4.8 cu. ft. in the case of Pocahontas coal. In other words to obtain like results at very high ratings the Mid-West coals require about two and one-half times as much furnace volume as the Eastern coals.

10 Stoker limitations may also be imposed by the arrangement of boiler baffling. For high overratings with Mid-West coals very large volumes of flue gas must be handled. Resistance to the passage of these gases through the setting increases as the square of their velocity. At high ratings therefore, the drop in pressure through the furnace and boiler increases rapidly, at 200 per cent of rating becoming four times the drop at 100 per cent rating, and at 300 per cent of rating becoming nine times the drop at 100 per cent rating. Draft pressures must therefore be adequate to overcome these high frictional losses or capacity will be limited. Location of baffles also, to give uniform velocities throughout the path of gas travel, is an extremely important phase of boiler and furnace design which, until recently, has had too little serious attention and has been a limiting factor in boiler efficiency.

11 The problem of ignition offers another limiting factor in the operation of stokers using Mid-West coals, especially at high overratings. Furnace design must permit of the concentration of sufficient heat at the point of entrance of the fuel to accomplish a rapid distillation and strong ignition of the hydrocarbons. The

[1] Bulletin No. 135 (1917).

necessity for, and the practice of, installing a properly designed reflecting arch to accomplish this is well known. The high volatile percentage of our Mid-West coals tends to facilitate ignition and is especially favorable when the volatilization may be accomplished rapidly and at comparatively low temperatures. This produces the greatest percentage of methane (CH_4), which is the simplest hydrocarbon and most easily consumed. Successful ignition also depends to some extent upon the size of the coal, smaller sizes igniting more quickly than larger lumps on account of the greater surface displayed to the heat.

EFFECT OF LIMITATIONS ON OPERATION

12 The effect of these various limitations in the case of chain-grate stokers utilizing natural draft is that maximum capacity is limited to the burning of from 40 to 45 lb. of coal per sq. ft. per hour, which may be somewhat increased at a large sacrifice in efficiency. Above this rating, however, the ash-pit loss increases excessively, and it is almost impossible to maintain ignition. In the case of forced-draft chain grates the limits as to rates of combustion are considerably higher, and greater flexibility of operation is also obtained through a more definite control of air supply. However, when the coal rate is increased beyond approximately 60 lb. per sq. ft. per hour, clinkers and slag complications make efficient operation difficult. In addition to this, temperature conditions at high ratings become such as to make furnace and brickwork maintenance a serious problem.

13 In the case of the overfeed type of stoker, operating under natural draft, a coal consumption somewhat less than that obtainable on the chain grate is generally recognized, this being approximately 30 lb. per sq. ft. per hour. Trouble from the avalanching of the fuel bed is reported beyond this rating, resulting in breaks and bare spots within the fuel bed and temporarily limiting both efficiency and capacity. In the application of forced draft to overfeed types of stoker, over-ratings may be somewhat extended. A test by the Bureau of Mines on a Murphy-type stoker developed a combustion rate of 60 lb. per sq. ft. per hour.

14 In the case of the underfeed stoker, particularly where equipped with multiple retorts and forced draft, a quick response may be had to all changes in load, resulting in an exceedingly flexible control. For continuous operation the maximum amount of coal which can be burned successfully seems to be from 900 to 1000 lb.

per retort per hour. For high overratings over short periods a combustion rate of 1400 lb. per retort per hour may be established. The limiting features as to capacity and efficiency in the underfeed stoker utilizing Mid-West coals lie mainly in the high ash content of the fuel and its low fusibility, which at high overratings result in clinker formation and sometimes in the fusing over of the entire surface of the fuel bed adjacent to the dump plates. While such conditions are being rectified combustion is partially suspended, capacity limited, and efficiency low.

15 In conclusion, it may be generally stated that the application of mechanical-stoking equipment to the use of Mid-West coals has proved to be eminently successful. The peculiar characteristics of the fuel have called for especially designed arrangements for air supply and for ignition. Furnace volumes and gas passes have also had scientific attention, with the result that the most difficult problems, namely, those incident to operation at high rates of combustion, have been satisfactorily solved. The particular type of stoking equipment best adapted to these coals may be specified as that one which most fully overcomes the many limitations to capacity and efficiency which have been found to exist.

No. 1788

CAPACITY AND EFFICIENCY LIMITATIONS OF STOKERS USING MID-WEST COALS

By John E. Wilson, Chicago, Ill.
Member of the Society

According to the author, less progress has been made in the direction of securing efficient combustion than in any other immediately connected with stoker service. In the case of natural-draft chain-grate stokers the greatest losses are those due to the dry chimney gases and the carbon in the ash, and the possibility of making a material gain in the efficiency lies to a great extent in the ability to reduce these two losses. A certain amount of air admitted over the fire has been found advantageous when high-volatile coals are used, such as those found in the Middle West.

Capacity is governed principally by the available draft and ratio of total boiler heating surface to the grate area. Chain-grate stokers provided with forced draft operate successfully under continuous high rating, respond quickly to load requirements, and will successfully burn coal containing a large percentage of ash which fuses at a comparatively low temperature.

Underfeed forced-draft stokers in the Mid-West districts have a large number of very satisfactory operating records to their credit both from an efficiency and a capacity standpoint. Some types of overfeed stokers have also been used with good success in this section where only a limited capacity was required and suitable coal was available.

The necessity of securing improved refractories as well as improving the furnace design is emphasized by the difficulties encountered with the brickwork when forced firing is employed, and it is possible that these may prove factors governing the capacity which will be obtained in future installations.

THE cost of stoker repairs per ton of coal burned depends to a large extent on the quantity of coal consumed per square foot of grate surface, or its equivalent, rather than on the type of stoker used. The average cost of repairs to a number of chain-grate and underfeed stoker installations has been found to be 3.7 cents and 5.4 cents, respectively, the cost of repairs to the fans and air ducts required by underfeed stokers being included in the latter figure. The chain-grate plants mentioned will burn, on an average, 30 lb.

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of dry coal per sq. ft. of grate surface per hour, while the underfeed-stoker plants will average 550 lb. per retort per hour.

2 In considering the repair and operating records from numerous plants, the fact has been brought out that the stoker is one of the most dependable pieces of equipment in the power plant. The stoker, up to the present time, has been primarily a labor-saving device. The importance of this feature has increased with wages, together with the uncertainty and scarcity of labor. The decision to install stokers in many plants has been governed to a large extent, if not entirely, by this factor. The saving in the cost of operation effected by stokers is frequently considered along with the saving made by the coal and ash-handling machinery. In this way the total saving often appears credited to the stokers alone. Since all or part of the coal-handling, and all of the ash-handling, equipment is applicable to a hand-fired plant, care should be taken to make a distinction between these figures. The quantity of steam required for driving stokers, also for draft fans, should in all cases be charged against stoker operation when considering these costs. The quantity of steam required for this purpose will vary from 0.3 per cent on stokers using natural draft to 3 per cent where induced or forced draft is used.

MECHANICAL STOKERS FROM THE STANDPOINT OF EFFICIENT COMBUSTION

3 Upon considering mechanical stokers from the standpoint of efficient combustion, it is apparent that less progress has been made in this direction than in any other immediately connected with stoker service. The saving due to the increased efficiency obtained with stokers over hand firing will cover only to a small extent the fixed charges on the increased investment required by the stokers. The different types of stokers will be taken up separately in considering the conditions relating to combustion efficiency and capacity.

4 Chain-grate stokers operating under natural or induced draft have been used extensively through the Middle West for a number of years. In supplying coal to the stoker magazine it is the usual practice to feed it from an overhead bunker through chutes. These chutes are either fan-shaped at the discharge end, or hung so as to allow sufficient swing to distribute the coal in the magazine. In gravitating down the chute the various-sized pieces of coal have a tendency to separate from one another. This action causes the

coal to be delivered in the magazine with the small coal in the center, increasing in size toward the ends. The result of this is a marked tendency for the fire to burn off unevenly. The use of overhead lorries for weighing coal to the stoker magazine eliminates this condition to a large extent.

5 A large percentage of the screenings, or slack coal, used on chain grates will pass a $\frac{1}{2}$-in. mesh screen. In some cases this fine coal has found its way to the sieving pit in such quantities as to represent 15 per cent of the total amount of coal fed to the grate. As a rule the sievings are handled in one of two ways: they are conveyed back to the main coal bunkers, or returned directly by hand to the stoker magazine. A comparatively small quantity of this coal on the grates will be sufficient to upset conditions in the furnace, resulting in lower capacity and efficiency. The stoker manufacturers have made attempts in some cases to reduce the amount of sievings by providing air openings running diagonally across the links, also by substituting skids in place of rollers. Efforts in this direction have been only partly successful.

6 The design and location of the furnace arch have an important bearing on the operation of a chain grate, both from an efficiency and capacity standpoint. It is highly important that the hydrocarbons driven off from the coal in the distillation zone at the front of the stoker be consumed in the furnace before having an opportunity to come in contact with the heating surface of the boiler. The ignition rate is governed principally by the design of the arch, and this affects the quantity of coal consumed to a large extent.

7 The following figures are those of a representative heat balance from boilers equipped with natural-draft chain grates burning Illinois coal:

	Per cent
Heat absorbed by water	70.0
Heat loss due to dry chimney gases	14.5
Heat loss due to combustible matter in the ash	5.5
Heat loss due to burning of hydrogen	5.0
Heat loss due to incomplete combustion of carbon	1.5
Heat loss due to moisture in coal	0.5
Heat loss due to moisture in air	0.5
Heat loss due to radiation and unaccounted for	2.5
	100.0

8 It will be seen from this that the greatest loss from any two combined sources is from the dry chimney gases and carbon in the ash.

This indicates that the possibility of making a material gain in the efficiency lies to a great extent in the ability to reduce these two losses. The importance of reducing the excess air is apparent, since some 40 per cent of the loss due to heat carried away by the dry stack gases results from the excess air admitted to the furnace. The most serious fault, perhaps, of the chain-grate stoker is the failure to provide adequate facilities for controlling the air supply to the furnace. Adjustable dampers have been installed on some natural-draft chain grates for this purpose, but the results obtained are only partly satisfactory. If attempt is made to burn the carbon out of the ash, it is found that an increased amount of excess air is admitted through the rear of the grate.

9 The short-circuiting of air through the well-burnt-out ash, in addition to carrying heat away from the boiler, tends to slow down the ignition and combustion rate on account of reduction in furnace temperature. On the other hand, if the fuel bed is carried well up to the water back, an excessive amount of carbon will be wasted in the ash. On account of the conditions just mentioned, after a certain point is reached the combined loss from stack gases and carbon in the ash remains approximately the same — a gain in one being offset by an additional loss in the other.

10 The baffle found on chain-grate stokers at the rear of the grate helps to prevent an excessive amount of air from finding its way into the furnace between the water back and grate. The condition which tends to discount the good to be derived from the baffle is the variation in size and nature of the coal, which makes it difficult to maintain a uniform length of fire extending up to the water back at all times. The stoker baffle is necessarily located in a place difficult of access, with the result that it is likely to be neglected.

11 Various experiments have proved that a certain amount of air admitted over the fire is an advantage where high-volatile coals are used, such as are found in the Middle West. This will be appreciated when consideration is given to the fact that practically no free oxygen passes through an active fuel bed of a uniform thickness which would be available for uniting with the combustible gases driven off of the coal soon after it passes into the furnace. With the present stoker design there is no doubt considerable air leaking in around the ledge plates, coal gate, etc., which furnishes in a more or less uncertain way some of the air required above the fuel bed.

12 The chain grate disposes of the ash very successfully, even from coals which run high in ash having a comparatively low fusing

point. This is a material advantage as it permits the use of high-ash coal, which is practically valueless on account of clinkers when used on other types of grates.

13 The various factors enumerated in the preceding paragraphs relate primarily to efficiency. Operating records from a large number of plants show that an actual operating overall boiler efficiency of 70 per cent is difficult to maintain. This covers plants where natural draft is used and boilers are not equipped with economizers.

LIMITATIONS GOVERNING CAPACITY OF STOKERS

14 The capacity is governed principally by the available draft and ratio of total boiler heating surface to the grate area. With a draft of 0.35 in. over the fire and a ratio of heating surface to grate area of 50 to 1 or less, a capacity of 170 per cent of rating should be obtained without any sacrifice in efficiency. When the ratio of heating surface to grate area is around 60 to 1, a capacity of 140 per cent rating is seldom exceeded. This is based on an available draft of 0.35 to 0.40 in. over the fire.

15 In many installations where the boilers are equipped for induced draft, ratings of 180 per cent to 190 per cent, and even higher, are obtained with chain grates. It is common practice to use economizers in connection with induced draft.

16 It has been pointed out in a number of instances that small plants burning from 50 to 150 tons of coal a day operate very economically as compared with larger plants in the same district. This is due in a large measure to the fact that the supply of coal for the small plant is obtained from one source. Under these conditions the furnace design may be changed to suit the coal used, and the most satisfactory method of operating the stokers can be determined and adhered to. The coal for the large plant, on the other hand, is secured from a number of different sources and varies materially in quality and other characteristics. This requires very close attention in order to make the numerous changes necessary in the feed, air supply, etc., to suit the different coals.

17 Several different makes of forced-draft link or chain-grate stokers have been placed on the market in the past few years. A number of installations of this type of stoker are to be found in the Middle West. In making provisions for furnishing air under pressure below the fuel bed, wind boxes and dampers were installed in such a way as to divide the fuel bed into several zones. The control of the

air supply to any zone can be regulated independently of the other sections. This feature makes it possible to admit the air to the fuel bed where it is required, and in the proper quantity. It follows that an increased efficiency is obtained, as the excess air is reduced to a minimum. The quantity of unburned carbon in the coal is also much less than found with other types of stokers. The total gain in efficiency, however, is not as great as might be expected from a careful consideration of the principle of air supply and regulation as shown on paper. The increased investment required by the forced-draft chain grate, together with the increased cost to operate, materially cuts down the saving in money due to the increased efficiency. The satisfactory operation of this type of stoker under continuous high rating and the quickness with which it responds to load requirements form a very important feature. It is possible to use coal containing a large percentage of ash which fuses at comparatively low temperatures without any difficulty on the forced-draft chain-grate stokers. This type of stoker at the present time is without doubt better adapted for burning the Mid-West coals economically at high ratings than any other stoker on the market.

18 Underfeed forced-draft stokers in the Mid-West districts have a large number of very satisfactory operating records to their credit, both from an efficiency and a capacity standpoint. The difficulties experienced with holes burning in a fuel bed, 6 or 7 in. thick, are eliminated on the underfeed. This is due to the method of feed and the greater thickness of the fuel bed on the grate. This type of stoker shares the advantage with other forced-draft stokers of being able to bring a boiler up to 250 to 300 per cent rating in a very short time. The nature and quantity of ash in the coal form probably the most important single item governing the successful operation of underfeed stokers. With a heavy fuel bed the tendency is for the ash to ball up and clinker, preventing the carbon from being burnt out in the furnace. This condition at the same time tends to interfere with the continuous operation of the stoker at high capacity. This is on account of the necessity, even with the so-called clinker grinder, of getting the clinker loosened up and into the ashpit by the use of bars in the side doors of the furnace. The ironwork at the rear of the stoker, under conditions just mentioned, comes in for very severe service, and the repairs thereto are likely to be excessive.

19 Some types of overfeed stokers have been used with good success in this section. One of the greatest drawbacks to the over-

feed stoker for present-day installations is its small capacity range. Where only a limited capacity is required, and suitable coal is available, results that will compare favorably with those from other stokers can be obtained.

20 The necessity of securing improved refractories, as well as improving the furnace design, is emphasized by the difficulties encountered with the brickwork when high boiler capacities are obtained. While this condition is probably in no way peculiar to stokers using coal from this section, it is possible that it may prove to be one of the factors governing the capacity which will be obtained in future installations.

21 It is natural for every stoker manufacturer and every engineer in charge of a steam plant to mention only the creditable results obtained with his particular equipment or plant. Due to this fact, together with the various systems used in keeping records of the operations, it is very difficult to obtain definite information which can be considered representative. As against the many records that can be produced to show that the figures mentioned herein are out of line, an equal number of records can be cited to show that they are representative of present-day stoker operation.

22 One condition of primary importance to the stoker manufacturer has come about in the past few years, and that is that the class of help in the boiler room has improved to such an extent that a piece of equipment is not barred from consideration on the ground that it requires intelligence to operate it.

No. 1789

GRAPHICAL METHODS

By W. C. Marshall, New York, N. Y.
Member of the Society

After pointing out the value of graphical methods and the wide use which may be made of them, the author calls attention to the pressing need for the standardization of diagram forms in order that comparisons of the results plotted by different investigators on the same subject may be facilitated. So important does he deem the subject that he advocates a course in every school, college and university covering the use and construction of graphical diagrams together with the theory of graphical methods. Various types of diagrams in frequent use are described in the paper and the advantages of the alignment form are brought out.

GRAPHICAL methods comprise all those methods of representing the relations of objects or facts by means of the relations between the lines of a diagram. All devices for representing by geometrical figures the numerical data which result from the quantitative investigation of phenomena are included under this title.

2 Graphical methods are employed to a large extent in physical investigations as aids to calculations and for the purpose of exhibiting the nature of the laws of variation of various phenomena. Their principal use is (*a*) to show the mutual variation of two quantities, as evidenced in the conveying of information, as when parallel lines of different lengths are exhibited which are proportional to the populations of different countries or to the populations of a given country at different periods of time; and (*b*) to aid in numerical or logical calculations, as when a curve is drawn through points whose coördinates represent the population of a country at successive decadal intervals and this curve is used to ascertain the population at other dates.

3 There are three classes of graphical methods: (1) those which make no use of the continuity of space except to show that the extremities of lines are connected (graphs); (2) those which use only the projective properties of space such as drawings and maps; and (3), those which use only the metrical properties of space and

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produce diagrams intended to be measured. To this last class belongs graphical statics.

4 Graphical methods of discussing experimental data are of great convenience and importance when the problem under investigation is to determine the law or fundamental relationship existing between two quantities. This type of problem arises frequently in technical and scientific investigations.

FUNCTIONS OF GRAPHICAL PRESENTATION

5 Graphical methods are inferior to numerical ones in accuracy. Ease and rapidity are essential when it is desired to compare many sets of facts, because if the mind is long delayed in taking in the facts of one set it loses count of the others. The function of graphical representation is to facilitate comparison. By tabulation we reduce facts to a logical order. By graphics we add to their value.

6 A graph is a pictorial representation or statement of a series of values all drawn to scale. It gives a mental picture of the results of statistical examination in one case, while in another it enables calculations to be made by drawing straight lines, or it indicates a change in quantity together with the rate of that change.

7 Diagrams do not add anything to the meaning of statistics, but when drawn and studied intelligently they bring to view the salient characteristics of groups and series and suggest in what directions investigation is needed. They are generally illustrations of the analysis obtained by reference to tabulated matter. They clarify the latter but do not displace it; in fact, their use supplements it and enables conclusions to be formed after a cursory inspection.

8 Simplicity and truthfulness should be the aim in graphical presentation. Find the best form of representation to use and seek to fit it to the requirements of the problem as to order of arrangement of details, spacing, size of figure, and methods of making emphatic the relations between facts. A skillful writer can often devise statistical diagrams of other kinds which help the visualization of a complex argument. The final test of a diagram's value is its legibility and clearness of meaning. The diagram should carry on its face a sufficient definition of the facts represented.

9 The psychology of statistical diagrams seems to depend on the difficulty of holding in the mind at one time all the mass of numerical facts contained in a series of statements, and for this

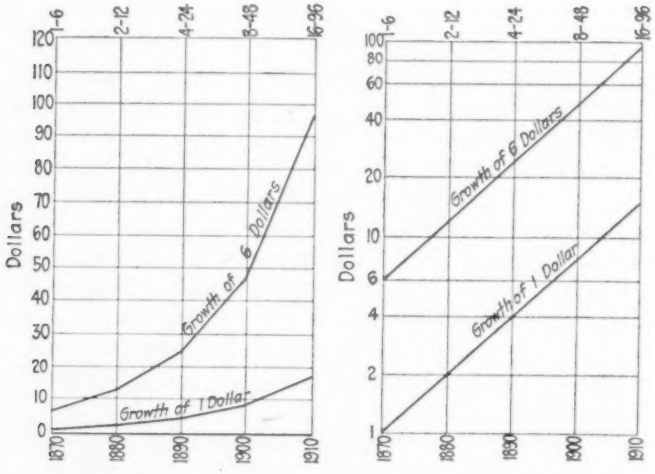
reason tables are used. Tables, however, compensate only in part for this fault, and it becomes necessary to still further crystallize the facts in diagrams.

NECESSITY FOR STANDARDIZATION OF DIAGRAM FORMS

10 What there should be in every school, college, and university in the country is a course covering the use and construction of graphical diagrams together with the theory of graphical methods. This course should cover all the various devices for enabling the eye to quickly grasp the features which are delineated on paper or board and useful in illustrating past, present, or future happenings. Efficient management cannot afford to do without graphical methods in planning, production, cost accounting, purchasing, sales, engineering, stores, or organization. At the present time there is a total lack of standardization of the form of diagram to use for nearly every kind of representation. This makes it difficult to compare the reports of different investigators on the same subject because their diagrams are not constructed alike. If one takes the vertical axis as the axis for r.p.m., another takes the horizontal axis, while a third uses rev. per sec. instead of r.p.m., which makes comparison almost impossible

11 Certain kinds of data are almost always represented by the same kind of diagram. Certain graphical recording instruments use charts of the same form, either circular or in ribbons, with the records on them made in the same way. The motion of nearly all of these charts is produced by clockwork, therefore one of the units is a time element, usually an hour, which may be subdivided as much as may be desired. The other unit of the chart may be temperature, pressure, volume, or action, but time is common to all and is comparable in all. The indicator card of a steam or internal-combustion engine has been made in a standard way for many years and comparisons of indicator cards are easily made for this reason. Diagrams which show the torque of gasoline motors have been standardized by the Society of Automotive Engineers, which requires them to be made out on a specified form. This enables graphical comparison to be made of engines tested in different laboratories as the test results are presented in the same manner. Standardization of other types of diagrams is needed and as soon as possible, but this cannot be accomplished without the coöperation of all those who are interested in their use.

12 Standardization gives the maker and reader of diagrams complete familiarity with the various graphic forms used. It facilitates the comparison of data displayed in different diagrams and simplifies the transfer of a plot from one sheet to another. It also aids in developing simplicity of form, which is a prime requisite of the graphical method. Readers of statistical diagrams should not be required to compare magnitudes in more than one dimension.



(a) Arithmetic Chart (b) Logarithmic Chart

FIG. 1 GROWTH OF 1 DOLLAR AND 6 DOLLARS WHEN PLACED AT COMPOUND INTEREST FOR 40 YEARS. TWO CHARTS COMPARED

Visual comparisons of areas are particularly inaccurate and should not be necessary in reading such diagrams.

THE RATIO TYPE OF DIAGRAM

13 The plotting of historical curves is often misleading as to comparative rates of growth of different things. If we were concerned with absolute amounts it would not be so bad, but we often wish to determine the percentage of increase or proportional rate of increase, that is, the ratio in which the increase during a year, for example, bears to the amount at the beginning of the year. It is also difficult for the eye to judge without artifical aid the ratio of increase at different parts of the same or different curves. One method is to make the distances on one of the scales represent the

logarithms of the amounts instead of the true amounts. By this method lines of equal slope denote equal rates of change and the curves become parallel straight lines. This is called the "ratio method" of plotting.

14 The ratio diagram has several advantages over the difference method, one being that a straight-line graph in a ratio plot means uniformity in percentage growth, while the same uniformity in a difference plot will be represented by an exponential curve.

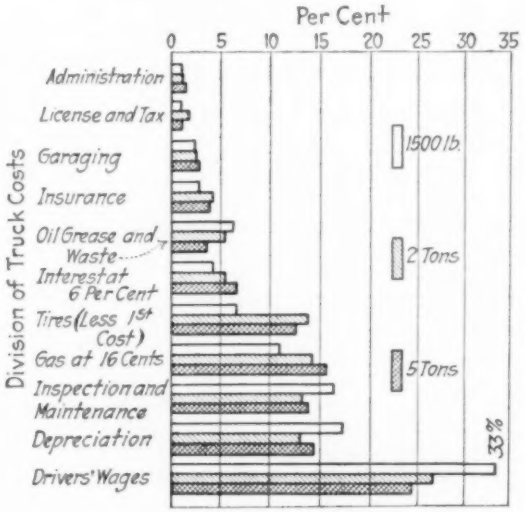


FIG. 2 DIVISION OF TRUCK OPERATING COSTS. COMPARISON OF THREE SIZES, 1500-LB., 2-TON AND 5-TON

The ends of such a curve are almost useless, but a ratio line is of the same value at all points. Equal rates of growth on a ratio diagram are clearly shown by parallel lines. We can move a curve bodily for comparison with another without impairing its value, which is not permissible on a difference diagram. An example of the false comparison of two curves by the difference method and by the ratio method is shown in Fig. 1. The growth of one dollar and of six dollars when placed at compound interest for forty years does not appear the same in (*a*) as it does in (*b*), although the percentage increase is the same and is so shown in the ratio diagram (*b*), where the lines of growth are parallel.

TYPES OF DIAGRAMS IN FREQUENT USE

15 Having dealt with the subject of graphical methods from an abstract point of view, a few concrete cases may now be considered. The type of diagram most frequently used to illustrate frequency and magnitude alone consists of lines and bars. Some use has also been made of surfaces and volumes, but this is not to be

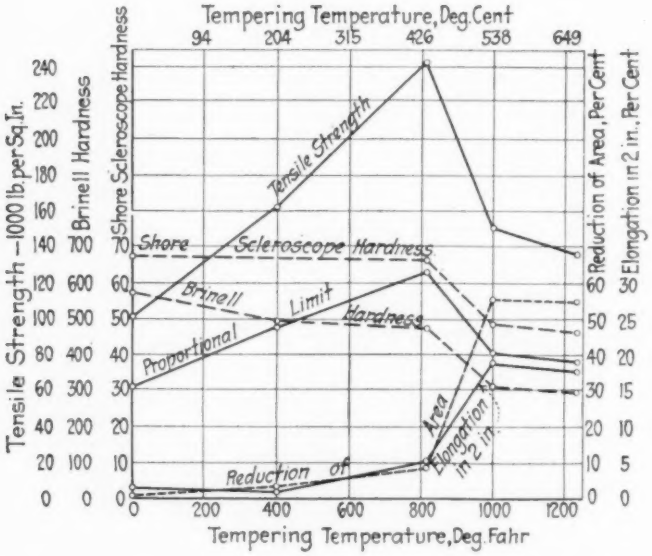


FIG. 3 EFFECT OF VARYING TEMPERATURES ON THE PHYSICAL PROPERTIES OF A HIGH-CHROMIUM STEEL, OIL-QUENCHED FROM 2100 DEG. FAHR.

recommended because of the difficulty of grasping more than one dimension by visual inspection.

16 A diagram of this type is shown in Fig. 2, the comparisons being made between different sizes of trucks as to the costs of the same items. It shows plainly how the size of truck effects certain costs more than others, and in a more convincing way than if it had been given in tabular form.

17 A graphical representation of the behavior of steel is shown in Fig. 3. This diagram gives all the information necessary to have in order to properly judge how this steel must be treated to obtain certain results. The upper and lower bases are a centigrade and

fahrenheit temperature-conversion scale. There are five scales for the six curves.

18 A type of diagram of the non-mathematical kind is that used for illustrating the route or path of an order from source to

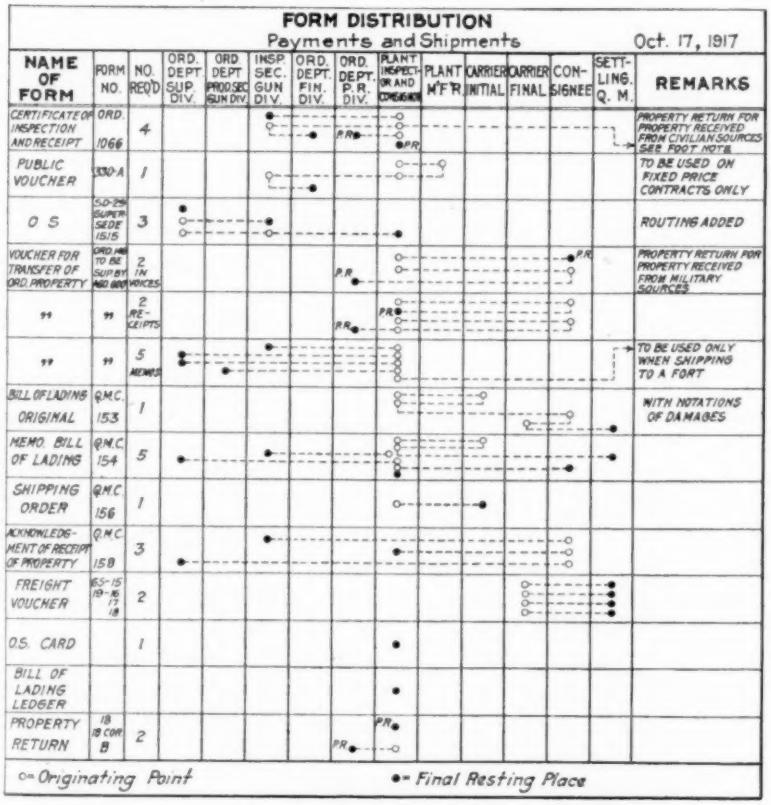


FIG. 4 DIAGRAM SHOWING METHOD OF HANDLING FORMS IN ORDNANCE DEPARTMENT, U.S.A., 1917

destination, or the path laid out for keeping track of an article of manufacture as it passes through the shop. An outline of the processes of manufacture or a flow sheet of all material obtained as a by-product during the reduction of raw material to final output can also be depicted by means of graphical methods. As an example of the first type of diagram Fig. 4 shows how certain forms in the Ordnance Department of the United States Army were handled

after they were made out. It shows the place where they were finally filed.

19 Organization diagrams, or charts as they are usually called, belong to the above type of diagram. Their function is to arrange,

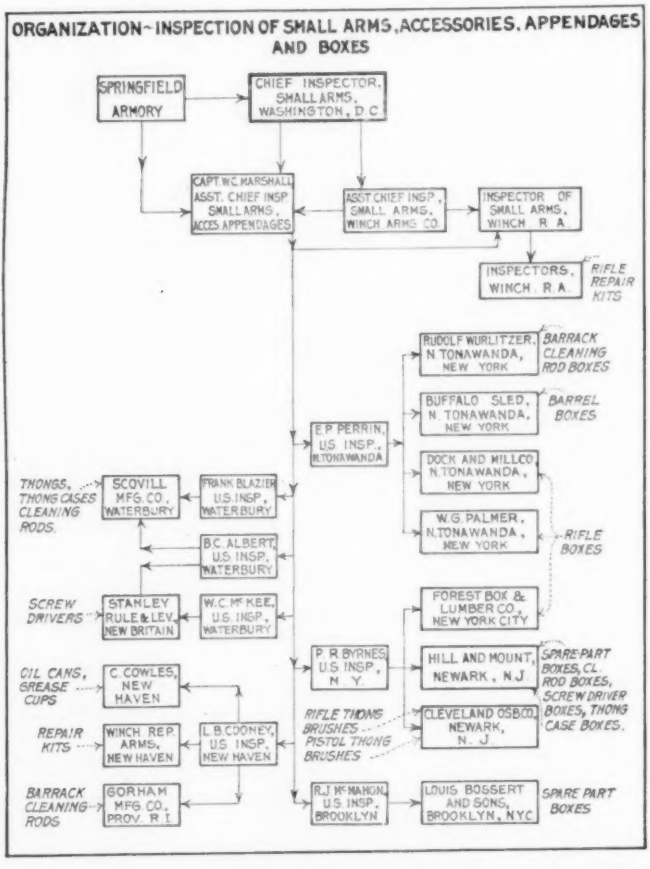


FIG. 5 ORGANIZATION CHART OF SMALL-ARMS ACCESSORY INSPECTIONS, ORDNANCE DEPARTMENT, U.S.A., 1917

in a clear form which can be quickly grasped by the executive, the plan of organization of the working forces of a business or corporation. They show each employee where he stands in the line of responsibility, who is his superior, who must report to him, and the relation of each one to the different branches of the organization.

These charts are made out in various ways by using circles or rectangles of different sizes to contain the names of the divisions and subdivisions of the organization, and often the names of the official who directs each one. The circles or rectangles are arranged according to the importance of the departments or positions, commencing with the president or board of directors and ending with the foremen or sub-foremen in the shops. The circles or rectangles are joined by lines in such a way as to indicate the relation of one department to another.

20 Not long ago the author had occasion to use a form of chart in connection with the duties of inspection of certain small-arms accessories in the Ordnance Department which differed somewhat from the general run of such charts and which is shown in Fig. 5. This chart shows at a glance the articles to be inspected, location of shop making them, and inspectors in charge of the work at each plant; also the routing of instructions from Washington and the Springfield Armory.

21 Plotting the results of experiments is usually done for the purpose of learning the law involved, as a line, straight or curved, can be drawn through the points plotted and its equation determined. The straight line as the representation of an equation finds its most important application in the determination of laws embodying the results of experiments. This is one of the most important applications of graphical methods. The curves of the conic sections are the most often found to satisfy the results of a series of experiments. Algebraic rules can be applied to the solution of simple or quadratic equations, but equations of higher degree or those not entirely algebraic can best be solved by graphs. In some cases no other method is possible.

ALIGNMENT DIAGRAMS

22 In most of the engineering periodicals the rectangular-coördinate diagram is used constantly and consists of a mass of squares, most of which are of no value whatever. The alignment diagram or "nomogram" has no lines on it except those marked with the scales. A coördinate diagram often has several lines plotted on it referred to several scales on its borders, while an alignment diagram represents but one formula and can be used for its solution alone.

23 As an example of the simplicity of the alignment diagram

when compared with the rectangular diagram, take the formula for the weight of rails:

$$W = 17^3\sqrt{(L + 0.0001LV^2)^2}$$

where W = weight of rails, lb. per yd.
L = greatest load on one driving wheel, tons
V = maximum velocity, miles per hour.

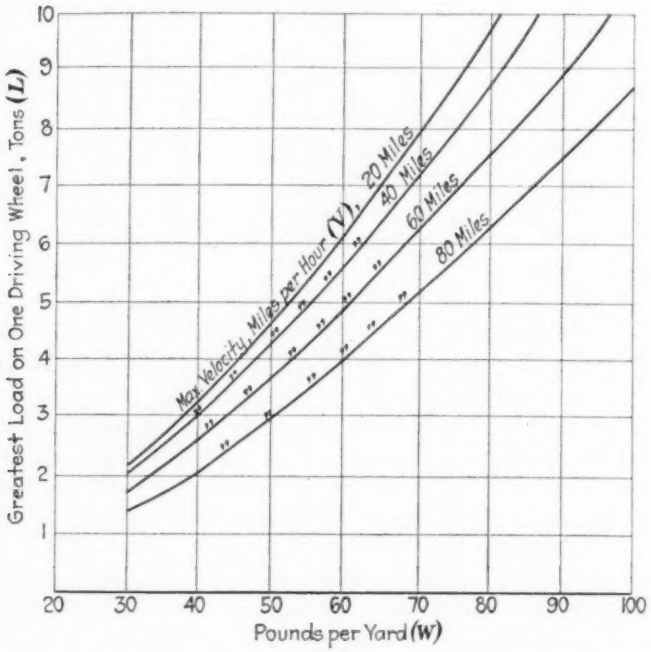


FIG. 6 DIAGRAM FOR FINDING THE WEIGHT OF RAILS PER YARD

The rectangular diagram for the equation is shown in Fig. 6 and requires much interpolation unless the V-curves are many, which makes the diagram hard to construct and to use. The alignment diagram for solving the same formula is shown in Fig. 7. Here three lines are drawn and divided into parts. The values of W for any values of L and V are found by placing a straight edge to connect the assumed values of L and V. The point where this edge intersects the W-line gives the answer.

24 The principles governing the construction of alignment diagrams were first formulated by Prof. Maurice d'Ocagne in his

work Traité de Nomographie, published in 1899. This has been followed by treatises of American writers, among them Prof. John B. Peddle and Prof. Joseph Lipka. These books cover the subject from a mathematical standpoint and should be consulted by those who wish to obtain an extended knowledge of the methods employed.

25 It is easy to read a nomogram with precision because of its few lines. It provides a tabulation of all possible values, enables

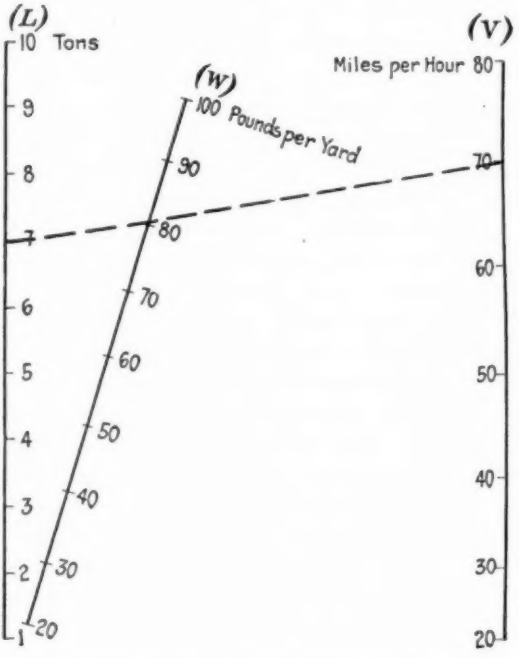


FIG. 7 ALIGNMENT DIAGRAM FOR WEIGHT OF RAILS PER YARD

solutions to be made irrespective of what quantity in the formula is unknown, and also enables one to observe instantly the effect of a change, either small or great, in any one of the variables. Equations with many unknown quantities, however, cannot be solved graphically without using higher mathematics.

RECORDING-INSTRUMENT DIAGRAMS

26 All of the diagrams that have been mentioned thus far have been those which were drawn by hand, using formulas or

tables as bases. Engineers and others, however, have found that it is very necessary in the keeping of records of many operations, processes or conditions, to have these records made by graphical recording instruments. These instruments record on paper occurrences which take place over a period of time. The periods may be from minute to minute, hour to hour, or day to day, but in nearly all cases the time element which is present necessitates the use of a clock as the driving mechanism for the paper. The paper may be a disk or ribbon on which the record is made by a marking point or pen connected to the operating device which moves in response to the varying condition to be recorded. Thus a recording thermometer marks the temperature, a combustion recorder the CO_2, a barograph the altitude, an accelerometer the acceleration, etc. A testing machine, on the other hand, does not depend on a clock for moving the paper as the operation of the machine rotates a drum on which the paper is wrapped. The same is true of the paper on a steam-engine indicator and on a recording dynamometer for testing the pull of a tractor. Such instruments serve an important purpose by making a record which can be inspected at any time and one which can easily be compared with any other, besides acting as a check on operations which involve the personal element.

27 The author is aware that nothing particularly new in the way of methods has been presented in this brief paper. He has endeavored rather to bring to the realization of the engineer the importance of "seeing things" and to impress on his mind the advisability of teaching the ideas and processes of graphical methods to the students in our schools, standardizing the art where it is possible, and using the methods in a business way.

DISCUSSION

Morris L. Cooke said that engineers were so accustomed to tables, blue prints and drawings that they did not appreciate the fact that most other people are uninterested in these things. Management engineers should study this problem in order to make charts attractive and easily understandable by the public at large; to popularize them so that the readers would be able to grasp the lesson conveyed by the chart. He spoke of an experience of his own in popularizing the reports of the Department of Public Works in Philadelphia by filling them with illustrations which would appeal to the average reader.

L. W. WALLACE spoke in confirmation of the remarks by Mr. Cooke and said that in his contact with industry and in the study of its problems, he had come to realize that not only management executives but the public at large had to be educated in many phases of engineering. Management engineers, therefore, should devote some consideration to developing means of portraying industrial and engineering information so that the public can read it intelligently and understand it. The graphical representation of facts in such a way as to catch and hold the eye and to convey the lesson intended is a step in the education of the public in industrial and engineering problems.

ROBERT T. KENT, who presented the paper in the absence of the author, spoke of the importance in constructing charts of selecting a unit to convey the desired meaning. He mentioned an incident in his own practice in which, having finally prevailed on the general manager to receive his performance reports in the form of charts, the latter wished the information in dollars as units. It was pointed out that this unit did not give the desired information, as the wages were going up and production per man going down, although the total production due to an increase in equipment and personnel was increasing. Cost in dollars did not mean much. The manager was finally prevailed upon to accept a graphic presentation of the information in payroll-hours per hundred pieces, which was a convenient unit. Thus the manager was able to correct influences which sent the curve up and to accelerate those which sent it down. Seeing the advantage of this form of presentation he then required that every activity of the shop be plotted in pay-roll hours or some other similar unit, leaving out the dollars.

H. WADE HIBBARD. One way of popularizing charts is to use colors, the different colors being used for different lines as the eye follows the colored lines easily. One should always assume in preparing charts that the user is ignorant of how to use it and should include on the chart an illustration of its use. This may be worked out for a specific example using dotted red lines which indicate the process to the user.

No. 1790

CAPACITY TESTS OF DRY-VACUUM PUMPS BY THE LOW-PRESSURE NOZZLE

By Snowden B. Redfield, Easton, Pa.
Member of the Society

The method of testing moderate-vacuum pumps described in this paper has a number of advantages. A nozzle is used in which the downstream or back pressure is higher than the critical. This permits the determination of several values of volumetric efficiency over a range of vacua without changing the nozzle. The method of plotting of results described by the author is a graphic proof of the accuracy of the theory of the nozzle and permits ready estimation of approximate performance of a given pump.

CAPACITY tests of dry-vacuum pumps are not common and their actual volumetric efficiency at a given vacuum is usually a matter of conjecture. Furthermore the characteristic curve illustrating the law of the change of volumetric efficiency with vacuum is but little known. It is the purpose of this paper to describe a simple, practical and undoubtedly accurate method of testing dry-vacuum pumps by means of the low-pressure nozzle and to show the characteristic curves of one kind of pump as manufactured by the Ingersoll-Rand Company. The pump illustrated and tested is but one of many similar all-plate-valve pumps tested by the writer. It is not built for extremely high vacua, but is intended for that extensive class of service met with in many industrial and chemical processes calling for what is known as a "moderate vaucum."

2 Although it is not always appreciated, it is a fact that a dry-vacuum pump is only a compressor working through a range of pressures low down on the scale. Every compressor is really a "booster"; that is, it takes in its air or gas at one pressure and compresses and delivers it at a higher pressure. The intake may be at any pressure: below atmosphere, exactly at atmosphere, or even a hundred or more pounds above atmosphere. Similarly the dis-

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charge pressure may be anything, as long as it is above the intake pressure. In all cases the cycle of intake, compression and discharge is essentially similar, although the weight of material and the power required will vary widely according to the pressures.

3 It is well to note here that the vacuum pump may have much the highest compression ratio to handle, although the actual pressures are the lowest. Thus, the compression ratio for a vacuum pump pulling 28.5 in. vacuum and discharging at atmosphere, is around 20; whereas for a compressor with 100 lb. discharge from atmospheric intake it is 7.8. A gas booster pumping from, say, 100 lb. intake to 400 lb. discharge has a compression ratio of only 3.62.

NOZZLE TESTS FOR AIR MEASUREMENT

4 Within the last ten years tests for capacity of air compressors by means of the low-pressure nozzle have become quite common, and the measurement of air has long since passed from the realm of assumption to that of practically standardized methods of determination with considerable accuracy. This being true of the air compressor, it at once becomes true for the dry-vacuum pump, as they have just been shown to be members of the same family, and by methods similar to those standardized for the air compressor we may obtain the exact amount of air handled by a dry-vacuum pump at any vacuum.

EFFECT OF CRITICAL PRESSURE RATIO

5 Before going further, a word should be said in explanation of the terms "low-pressure nozzle" and "high-pressure nozzle." These are the popular terms to designate whether the back pressure into which the nozzle is discharging is above the critical pressure or below it, respectively. The critical back pressure for air is 0.53 of the upstream, or initial, pressure, and if the back pressure is above the critical, the expansion ratio is relatively low, so the nozzle may be called "low-pressure." If on the contrary, the back pressure is less than the critical, the total expansion ratio is relatively high, and the nozzle is said to be a "high-pressure" nozzle. Table 1 gives the class of nozzle for various pressure ratios, assuming a 30-in. barometer.

6 As explained in many books on thermodynamics, if the back pressure is below the critical the expansion in the nozzle is only

down to the critical pressure in the throat of the nozzle and the rest of the expansion takes place beyond the throat without increasing the quantity of discharge. This immediately fixes the constant quantity of air a nozzle of a given size will pass from a given upstream pressure with any and all back pressures below the critical. For testing compressors with discharge pressures above 13 lb. gage, the nozzle discharging to atmosphere, different quantities of air

TABLE 1 DESIGNATIONS OF NOZZLES FOR VARIOUS PRESSURE RATIOS

	Upstream pressure		Back pressure		Ratio of back pressure to upstream pressure	Kind of nozzle
	Gage	Abs.	Gage	Abs.		
For Pressure Work	100	114.73	atm.	14.73	0.128	High-pressure
	20	34.73	atm.	14.73	0.424	High-pressure
	12	26.73	atm.	14.73	0.551	Low-pressure
	2	16.73	atm.	14.73	0.880	Low-pressure
For Vacuum Work	Atm.	14.73	2″ vac.	13.75	0.933	Low-pressure
	Atm.	14.73	8″ vac.	10.80	0.733	Low-pressure
	Atm.	14.73	15″ vac.	7.36	0.500	High-pressure
	Atm.	14.73	26″ vac.	1.964	0.133	High-pressure
	Atm.	14.73	28″ vac.	0.982	0.067	High-pressure

may be made to flow through a "high-pressure" nozzle of a given size by changing the upstream pressure. This can be accomplished by maintaining the compressor discharge pressure constant in a receiver by means of a throttle which passes the air to the nozzle at some slightly lower pressure. The lower pressure beyond the throttle, at nozzle entrance, will adjust itself automatically to the quantity of air to be passed, so that considerable flexibility may be obtained for compressor testing with this class of nozzle.

7 Due to the excessive noise of a "high-pressure" nozzle when discharging into atmosphere as in compressor testing, and the inaccuracies and difficulty of reading pressure gages of the spring type, the low-pressure nozzle with water-column pressure gage has been adopted by most of the leading compressor builders and is being incorporated into the new test codes of the Society now under revision.

8 Recently some capacity tests have been made upon vacuum pumps using a high-pressure nozzle. This is done by placing a nozzle in the wall of the vacuum vessel, drawing air from the atmosphere, and calculating the flow from well-known formulas. While this has been an assistance in some cases, it is not a flexible method and a different size of nozzle must be used for each vacuum at a given pump speed. This makes it a practical impossibility to test a vacuum pump at a specified vacuum, because one cannot tell just what size of nozzle will be required. With the low-pressure nozzle a given nozzle diameter may be used for a very considerable range of vacua, and a capacity measurement may be made at any specific vacuum desired.

9 In testing a compressor discharging at any pressure above atmosphere by a low-pressure nozzle, the air passes to a receiver where a throttle outlet holds the pressure at the desired point. This may be anything up to several thousand pounds, if desired. The air passing the throttle goes to a second receiver which is equipped with a nozzle of such size that it will pass all of the air from the compressor to the atmosphere with an upstream pressure less than 13 lb. Keeping below this maximum puts the nozzle in the low-pressure class, and many experimenters, including the writer, prefer to keep the nozzle pressure down to a few inches of water column. By keeping between $3\frac{1}{2}$ in. and 18 in. of water column, very simple empirical formulas may be used to calculate air quantity. Apparatus shown in Fig. 1, but connected to the discharge side of the compressor, can be used for measuring compressor capacity by reversing the low-pressure nozzle.

10 Coming back to vacuum pumps, inspection of Table 1 already given shows that as soon as the vacuum exceeds about 14 in. of mercury, a nozzle discharging from atmosphere into this vacuum is discharging into a pressure below the critical and we have at once a nozzle of the high-pressure class. We have already shown that a high-pressure nozzle of a given size discharging from a constant upstream pressure into a back pressure below the critical will pass only a constant air quantity, irrespective of how low the back pressure may become, and as in this case the upstream pressure remains atmospheric, the quantity of air passed will be constant for a given size of nozzle for all vacua above 14 in. It then follows that the vacuum obtained with a given size of nozzle of the high-pressure type, which passes a fixed quantity of air, can be changed only by changing the speed of the pump. If the pump speed is

increased, the air must expand after it has passed the nozzle, in order to fill the increased piston displacement, and this expansion must be accompanied by a reduction of pressure, which is the same thing as an increase in vacuum. Reduction of pump speed is accompanied by the reverse process and the vacuum will be decreased; but the fact remains that a high-pressure nozzle of a given size will pass only a constant quantity (weight) of air and will produce only one vacuum at a given pump speed. Furthermore it is impossible to select a nozzle of this type to test a given pump at a specified vacuum.

APPLYING LOW-PRESSURE NOZZLE TO VACUUM PUMPS

11 As a more flexible and probably more accurate method of testing dry-vacuum pumps, the set-up shown in Fig. 1 lends itself well. In this figure the vacuum pump is drawing from tank

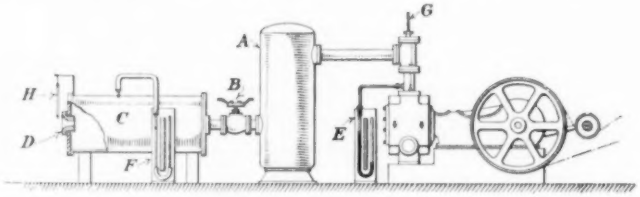


FIG. 1 APPARATUS FOR LOW-PRESSURE NOZZLE TEST OF VACUUM PUMP

A, and any desired vacuum may be produced in *A* and the pipe connecting it to the pump by adjusting throttle *B*. Throttle *B* is a globe valve provided with a long handle for close adjustment. Its stuffing-box gland is liberally daubed with heavy oil to insure against leakage. The running vacuum is observed on mercury column *E*, and, as said, this vacuum may be adjusted by *B* to anything desired. Tank *C* has in its end the nozzle *D*, taking air from atmosphere, and the pressure drop through *D* is shown by the water column *F*. A thermometer *G* shows the temperature of intake to vacuum-pump cylinder and thermometer *H* shows the temperature of the air entering the nozzle.

12 We now see that the apparatus is composed of three parts: the pump to be tested; tank *A* and piping in which any desired vacuum is to be maintained, measured by mercury column *E* and at temperature of thermometer *G*; and tank *C*, which by means of nozzle *D*, water column *F* and thermometer *H*, measures the amount of air handled. As *B* is adjusted, the quantity of air

handled will change. This will change the vacuum as shown on gage E, and water column F will also change to measure the new quantity of air passing through nozzle D. As the drop through D is small, the pressure in C is well above the critical at all times and the quantity of air passed will change with the slight vacuum in C, as measured by water column F. Thus there is a wide range of air quantities which can be made to pass through a low-pressure nozzle of a given size, and by proper adjustment of throttle B any number of test points may be obtained as close together or as far apart as desired, independent of the speed of the pump. Furthermore, any specified vacuum may be secured at will, up to the limit of the machine.

FORMULAS FOR CALCULATING FLOW QUANTITIES

13 For calculating the flow of a gas through an orifice or nozzle, the best-known formula for the case where the back pressure exceeds the critical is that which equates the energy on the two sides of the nozzle and assumes adiabatic expansion in the nozzle. This formula is given in all books on thermodynamics and most handbooks. As given by R. J. Durley,[1] it takes the form —

$$W = A\sqrt{2g\frac{\gamma}{\gamma-1}\times\frac{P_1}{V_1}\left[\left(\frac{P_2}{P_1}\right)^{\frac{2}{\gamma}}-\left(\frac{P_2}{P_1}\right)^{\frac{\gamma+1}{\gamma}}\right]} \quad . \ . \ . \ [1]$$

where W = weight of gas discharged per second, lb.
A = area of cross-section of jet, sq. ft.
P_1 = pressure on upstream side of orifice, lb. per sq. ft.
P_2 = pressure on downstream side, lb. per sq. ft.
V_1 = specific volume of gas on upstream side of orifice, cu. ft. per lb.
γ = ratio of specific heat at constant pressure to that at constant volume.

14 By transposition and substituting 1.406 for γ and $53.4T/P_1$ for V_1 we obtain a workable formula for air: namely,

$$W = \frac{2.04\,AP_1}{\sqrt{T}}\sqrt{\left(\frac{P_2}{P_1}\right)^{1.425}-\left(\frac{P_2}{P_1}\right)^{1.712}} \quad . \ . \ . \ . \ . \ [2]$$

where T is the absolute temperature (deg. fahr.) on the upstream side of the nozzle.

[1] Trans. Am. Soc. M. E., vol. 27, p. 195.

15 This formula assumes frictionless flow and for practice a coefficient must be applied. This coefficient for a nozzle having a well-rounded entrance should be 0.97 for diameters below 1 in., 0.98 between 1 in. and 3 in., and 0.99 for nozzles over 3 in. in diameter. With larger nozzles the coefficient will remain very close to 0.99. This coefficient has been very carefully determined by Dr. Sanford A. Moss, as described in his paper on The Impact Tube.[1]

16 As P_1 is the upstream pressure and P_2 the downstream, the expression P_2/P_1 is less than unity, and to raise this to a power is an awkward job. Also, as P_1 and P_2 are very nearly alike with a low water-column pressure, it is necessary to get the ratio to at least five places of decimals. Unless this is done, the difference of the powers will not be correct and the air quantity will not be exact.

17 To avoid using numbers less than unity, it is more convenient to call $P_1/P_2 = R$, which is greater than unity; then raise the R's to the required powers and subtract their reciprocals. By this procedure we obtain a more convenient formula, namely,

$$W = \frac{2.04\,CAP_1}{\sqrt{T}}\sqrt{\left(\frac{1}{R}\right)^{1.425} - \left(\frac{1}{R}\right)^{1.712}} \quad \ldots\ldots \quad [3]$$

where C = coefficient = 0.97 to 0.99

R = ratio of absolute pressure on upstream side of nozzle to absolute pressure on downstream side of nozzle, $= P_1/P_2$, expressed to at least five significant figures.

18 Even with this modification it must be admitted that the formula is cumbersome, and for that reason the writer plotted the curve shown in Fig. 2. For practical use, this should be accurately calculated and replotted to a large scale, capable of reading to five significant figures. The abscissæ are values of $R = P_1/P_2$, which are therefore greater than unity, and the ordinates represent values of the complete radical $\sqrt{(1/R)^{1.425} - (1/R)^{1.712}}$. By means of this curve the process of figuring the weight of air handled is very greatly simplified.

19 After obtaining the weight per second, or per minute, of the air flowing, this should be changed to the volume handled by the pump so as to express it as a percentage of the piston displacement. To be perfectly logical, it should be expressed in cubic feet at the temperature of pump-cylinder intake, as shown by ther-

[1] Trans. Am. Soc. M. E., vol. 38, p. 761.

mometer G, Fig. 1, and at the pressure of the cylinder intake, which is the vacuum shown by mercury column E. The cubic feet at this condition will be obtained by dividing the weight of flow calculated

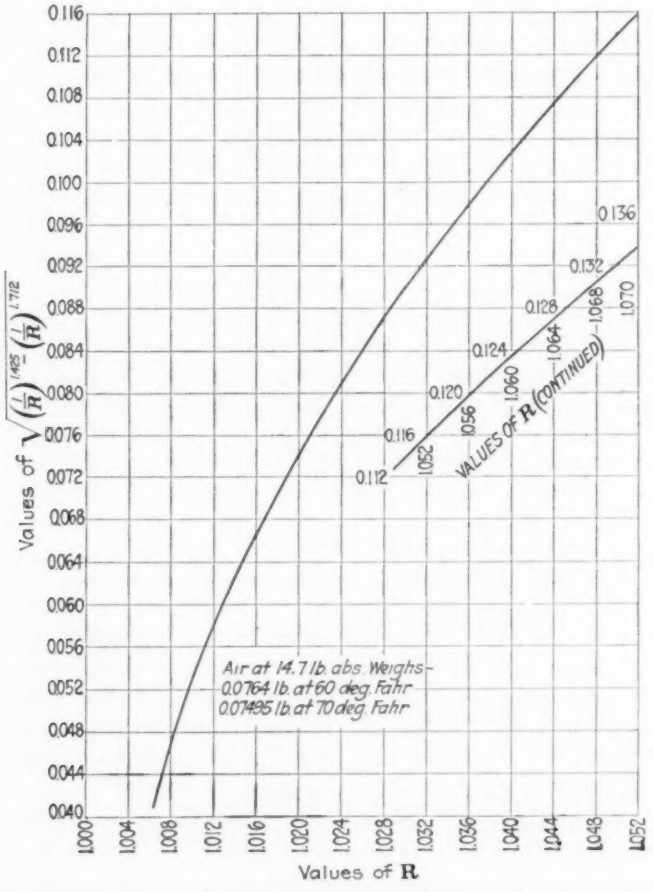


FIG. 2 FACTORS FOR CALCULATION OF AIR FLOW BY RATIONAL OR ENERGY FORMULA

from the nozzle readings — water column F and temperature at H, Fig. 1 — by the weight per cubic foot of air at the absolute pressure of vacuum read on U-tube at E and temperature measured at G, Fig. 1. The specific weight of dry air at any temperature may be calculated from the formula —

$$w = 0.1874 \frac{P}{T} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [4]$$

where w = specific weight at the pressure P and the temperature T, lb. per cu. ft.

P = absolute pressure at which the specific weight is to be found, lb. per sq. ft.

T = absolute temperature (deg. fahr.) at which the specific weight is to be found.

SLIDE-RULE FORMULA

20 For those preferring to use a slide rule for calculation, the following formula, based upon the principles of hydraulic flow, but modified empirically, will give values very close to the longer formula already given, provided the nozzle water column is not over 18 to 20 in. Below this limit the agreement is quite close at most points, but varying about ½ of 1 per cent, being that much high around 8 or 10 in. and that much low around 20 in. water column. The formula is so easy to use, however, that it is preferable to the longer one, and at low water columns — around 4 in. — it is certainly more accurate, as the ratios in the longer formula then become too small for accurate calculation. The formula is —

$$Q = 3.64\, Cd^2 \sqrt{\frac{HT}{P_m}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [5]$$

where Q = cubic feet flowing per minute at absolute pressure of *downstream* side of nozzle and temperature shown by thermometer on *upstream* side of nozzle

C = coefficient of flow = 0.97 to 0.99 as before

d = diameter of nozzle throat (smallest section), in.

H = observed nozzle water column, in.

T = absolute temperature (deg. fahr.) shown by thermometer on upstream side of nozzle

P_m = absolute mean pressure; that is, the average between the pressures on upstream and downstream sides of nozzle, lb. per sq. in.

21 As Q will be expressed at pressure of downstream side of nozzle and temperature shown by nozzle thermometer on upstream side, it must be recalculated into volume at the vacuum shown by mercury column at E and temperature shown by thermometer at G, Fig. 1, as before. This is of course done by changing the volume

inversely as the absolute pressures and directly as absolute temperatures. It should be noted that this formula expresses Q at downstream nozzle pressure, which is below atmospheric pressure by the amount of the nozzle water column observed.

PLOTTING TEST RESULTS

22 So much for methods of test and calculation. The next important step is the plotting of the results, and here we meet a fact which is very convenient for the experimenter. This fact is, that if the quantities of air at various vacua handled by a pump at constant speed are calculated into volumes at *atmospheric* pressure and the temperature of the intake to the pump cylinder (thermometer G, Fig. 1) and if these are expressed as percentages of the elapsed piston displacement of the pump cylinder, we obtain what may be termed "atmospheric volumetric efficiency," and this will be very nearly a straight line when plotted against vacuum. In Fig. 3 is such a curve, as shown by the diagonal line marked Atmospheric Volumetric Efficiency. This was taken from a small pump designed for moderate vacuum. In this diagram abscissæ represent vacua in per cent of the observed barometer and ordinates represent percentage volumetric efficiency.

23 Having arrived at this very simple form of volumetric efficiency curve, the ease with which points along the entire line may be read will be noted. This should be compared with the difficulty of reading the true volumetric efficiency curve, expressed at conditions of pump-cylinder intake, especially at the higher-vacuum points. In this part of the curve the line becomes so steep that close reading is very difficult and correct plotting is almost impossible. The slightest deviation to the right or left in drawing the high-vacuum part of the curve may make a difference of several per cent.

24 It should be noted that the volumetric efficiencies shown by both lines in the diagram, Fig. 3, are expressed at the temperature of pump-cylinder intake; that is, by thermometer at G, Fig. 1. This is correct because this is the temperature at which the pump receives the air. The only difference between the two curves is the pressure at which the air volumes are expressed. The temperatures being the same, the volumes are inversely as the absolute pressures, and as one of the pressures is atmospheric and the other the absolute pressure of the vacuum, the pressure ratio is simply the ratio of compression of the pump cylinder, provided the discharge

is atmospheric. Therefore, with atmospheric discharge and having calculated and plotted the atmospheric volumetric efficiency, as

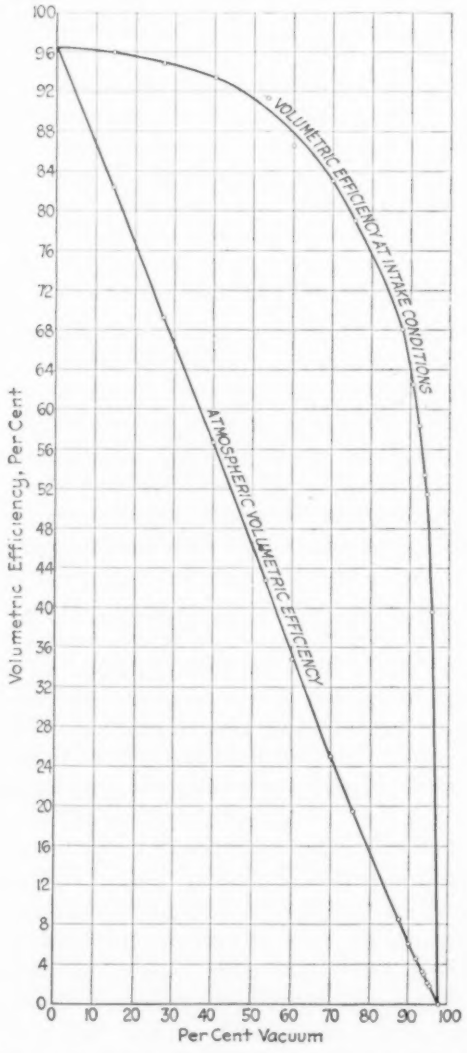


FIG. 3 METHOD OF PLOTTING VOLUMETRIC RESULTS

shown by the diagonal, almost straight line, the curved line of true intake volumetric efficiency may be obtained by multiplying the

values on the diagonal line by the ratio of compression. Of course, if the pump discharge is not atmospheric the values of the diagonal line should be multiplied by the ratio of the atmospheric pressure (barometer) to the absolute pressure of the vacuum at pump-cylinder intake.

25 Even after the true volumetric-efficiency curve is plotted, it is usually easier and more accurate to obtain the true values by picking them from the diagonal line and multiplying these values by the pressure ratios. The steepness of the curve of true volumetric efficiency is too great to read accurately.

EVIDENCE OF ACCURACY OF NOZZLE TESTS

26 Right here is an excellent opportunity to say a few words to convince the skeptical of the accuracy of the low-pressure nozzle as an instrument for the measurement of air, as exemplified by these vacuum tests. In such a test we have at the lower right end of the diagonal line the point showing maximum vacuum with closed intake, that is, with no air handled and no nozzle at all. Following up the diagonal line, the next four points were obtained with a $\frac{5}{8}$-in. diameter nozzle with water columns progressing from just under 4 in. to about 13 in. The next three points above these were with a 1-in. nozzle with water columns from approximately $3\frac{3}{4}$ in. to about $12\frac{1}{2}$ in. After these points follow nozzles of diameters 1.75 in., 2.00 in., and 3 in. with water columns up to just under 15 in., each nozzle being able to handle several different quantities. In such a vacuum test, then, we always have a series of nozzles beginning, in this case, with $\frac{5}{8}$ in. diameter and ending with 3 in. diameter, with water columns of several heights for each one; and in a very large number of tests which the writer has made, never has there appeared to be any visible break in the curve except what was well within probable experimental errors, showing where one nozzle size left off and the next size began. Considering that each smaller nozzle leaves off with a relatively high water column and the next larger diameter begins with a low water column, and furthermore that the smallest nozzle links up with the point of no nozzle and no air, this is also a very good argument that the formula used for calculating the quantity of air expresses the law of nozzle action very well indeed.

27 Another point should be noted, and that is that in such vacuum-pump tests the upper left end of the diagonal line, which

by the way, is also the upper left end of the true volumetric-efficiency line (the pressure ratio being unity), shows volumetric efficiency around 95 and 96 per cent. As at this point the machine is taking in air with atmospheric intake and also putting it out at atmospheric pressure, under these conditions the pump cylinder really acts as a rough kind of displacement meter checking the nozzle. Of course, with the extra large amount of air being handled under these abnormal conditions, a slight compression takes place inside the cylinder, producing a little heating and other small losses which prevent the volumetric efficiency from actually reaching 100 per cent of the cylinder volume. If the nozzle showed over 100 per cent volumetric efficiency we would know the nozzle was inaccurate. So far the writer has never seen a case of this sort, which cannot be said of some other methods of test he has observed and results he has seen reported in all seriousness. Being just under 100 per cent, and sufficiently under to be justified by working conditions, appears to the writer to be a very good argument for those who are skeptical. Other more scientific determinations, notably those of Dr. Moss, prove the result beyond dispute; but these vacuum-pump test curves have always served to quiet the objections of those who lack scientific knowledge but who require conviction by an obvious though approximate method.

APPROXIMATE COMMERCIAL ESTIMATIONS

28 With the characteristic of the diagonal line of Atmospheric Volumetric Efficiency we have a rather convenient method of approximately estimating the probable capacity of a given dry-vacuum pump under working conditions. If it is possible to close a valve in the intake of the pump so as to pull a "dead end" vacuum and compare this with the barometer this will give the point of the lower right-hand end of the diagonal line. With a pump of good quality the upper left end of the line, corresponding to the point of no vacuum or open intake and discharge, may be assumed at between 90 and 97 per cent volumetric efficiency. Between these two points the diagonal line may be drawn. It should probably sag just a little below a perfectly straight line, to represent approximately the characteristic of Atmospheric Volumetric Efficiency. From this line one may pick the probable volumetric efficiency corresponding to the vacuum observed under working conditions and apply this value to any calculations desired. It should be remembered that the result is expressed in air at atmospheric pressure.

29 It must be distinctly understood that such a method of estimating as just outlined cannot serve as a test of the pump. There are other factors, particularly valve and piston leakage, which, as described later, will modify the characteristic line; but it often happens that the engineer in charge of a power plant wishes to estimate the air leakage into his vacuum system. By observations of temperature and vacuum he can obtain the partial air pressure and by the method here described he can estimate the total volume of air and vapor mixture handled by the pump, from which he can arrive at a rough approximation of the air leakage existing. This method of rough estimation may also be applied to pumps handling air for exhaustion purposes or gases of various kinds. In fact, there are many interesting and valuable commercial calculations which can be made much more accurately when the approximate volumetric efficiency of the vacuum pump can be estimated.

EFFECT OF VALVE AND PISTON LEAKAGE

30 The effect of valve leakage has been mentioned as a cause for the departure of the Atmospheric Volumetric Efficiency curve from the straight line. It has been observed that with leaky valves the diagonal line will sag, especially at the high-vacuum, or right-hand lower end. With appreciable valve leakage the line coming up and to the left from the closed-intake, maximum-vacuum point, will have a very decided bend in the first two inches of vacuum decrease. This bend in the line will be curved downward, which of course puts the curve lower for a given vacuum. When the points on this sagging diagonal line are multiplied by the ratio of pressures, as described, to obtain the true Intake Volumetric Efficiency, this latter curve will simply lean further to the left. This means that the true volumetric-efficiency curve will not rise as rapidly as the curve corresponding to a diagonal atmospheric volumetric-efficiency line without sag. This again simply means that the true volumetric efficiency at a given vacuum will be lower, which naturally follows for a pump having valve and piston leakage.

31 With the above-described effect of leakage, it is clear that the true curves of a given pump cannot be drawn without actual nozzle-test measurements at several vacuum points near the maximum, as already described and illustrated in Fig. 3. For this reason any estimates based upon an assumed characteristic curve cannot be considered accurate, and if it is desired to be on the "safe side"

without actual capacity tests, it is well to put an appreciable sag in the diagonal line used as a basis of estimation. This refers particularly to the upper two inches of vacuum.

NOZZLE SHAPES

32 Something must now be said about the shape of the nozzles used. Coefficients already mentioned were for nozzles having well-rounded approach surfaces. With nozzles of this kind the coefficient has been found to be much more stable than when an attempt is made to use a so-called "sharp-edged" nozzle. It is a practical impossibility to make a nozzle with a perfectly sharp edge and it has been found that with the slightest degree of bevel or

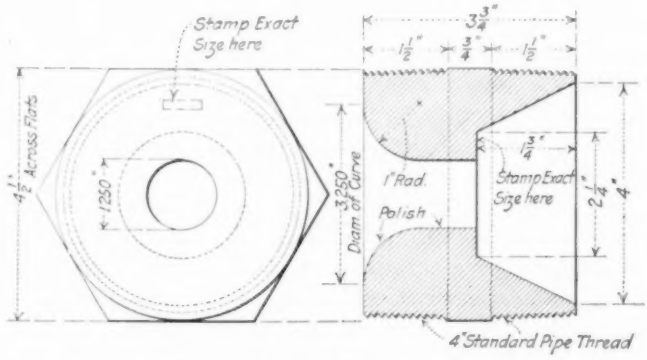


FIG. 4 CONVENIENT FORM OF AIR-MEASURING NOZZLE

rounding or other deviation from the perfect edge, the coefficient may vary widely. With the nozzle having a rounded approach it has been found that the coefficient will not change with change of approach radius within wide limits.

33 Fig. 4 shows a very convenient form of nozzle. It is made with a pipe thread on both ends so that it can be used at will on the intake side of a vacuum pump or on the discharge side of a compressor. The approach surfaces may have a radius of anything from 0.6 of the throat diameter upward, with inappreciable effect upon the coefficient. The throat is followed by a straight or cylindrical portion. It is recommended that this be not less than one-fourth the throat diameter, and it may probably be as long as the diameter without affecting the coefficient. Anything longer might bring in an additional friction influence to reduce the coefficient very slightly.

A short straight portion has been found really essential to secure smooth flow. The approved shapes of air-measuring nozzles will be described in the forthcoming air-compressor test codes now in preparation by the Power Test Codes Committee, and the latitude to be allowed will be quite broad, just as long as the approach curve is well rounded. Special curves of compound radii have not been found to possess any particular virtue. All this is good news to the experimenter along these lines.

NOZZLE ON DISCHARGE SIDE

34 In some cases it may be desirable to put the nozzle on the discharge side of the vacuum pump. There are two objections to this, although they are not very serious. First, to eliminate the pulsations of the discharge, two tanks will have to be put in series, with the nozzle in the second tank discharging to atmosphere. This aggravates the second objection, namely, the compression ratio, which would necessarily be increased, and therefore reduces the volumetric efficiency of the pump. This back pressure will also be different for different vacua and nozzle sizes. Any throttling between the two tanks to remove pulsation makes this much worse, and if the pulsations are not removed the readings of the nozzle will be incorrect. No throttling or otherwise damping of the pulsations of the nozzle water column will help, as the error comes in the pulsations of the actual air passing the nozzle. Throttling on the intake side is just what is required to get the vacuum, so that, all things considered, the nozzle on the suction side is undoubtedly to be preferred.

DISCUSSION

SANFORD A. MOSS. Reciprocating compressor work has often been very inaccurately done, and it is a pleasure to go over the author's careful methods. His explanations of the low-pressure and high-pressure nozzle systems are particularly to be commended. As he states, the theoretical formula by his Equation [**1**] should never be used in computation work. Equation [**5**] is, as he states, more accurate for most cases. The writer is of the opinion that it is slightly more accurate to use the downstream pressure P_2 instead of the mean pressure P_m in this formula. The author states that the results given by Equation [**5**] become slightly too low as the pressure differences increase. The use of P_2 instead of P_m allows

for this. As the author states, it is desirable to express cubic feet of air in terms of absolute temperature T_1 and pressure P_1 at the pump or compressor inlet. The general Equation [5] then becomes

$$3.642cd^2 \frac{T_4}{P_4}\sqrt{\frac{P_2H}{T}}$$

This formula gives directly the "cubic feet of inlet air" in one operation with good accuracy.

The author's empirical rule for volumetric efficiency is certainly very important. It seems to the writer that this would apply to any form of reciprocating compressor, as these machines are, as the author shows, all in the same family with his vacuum pump. In using the diagram of Fig. 3 for any reciprocating compressor, it is presumed that the abscissae would be the ratio of pressure rise to final pressure, or one minus the pressure ratio. Has the author, in his extensive experience with reciprocating compressors in general, found such a law to hold?

It is to be noted that such a law has semi-rational basis which may be demonstrated as follows:

Let P_4 = Inital pressure in the reciprocating compressor. This is the absolute pressure of the vacuum in the author's case.

P_3 = Final pressure. This is atmospheric pressure in the author's case.

C = Clearance volume as a decimal fraction of displacement

D = Volumetric efficiency, taking account of all losses except clearance

y_1 = Volumetric efficiency, with no pressure rise, or intercept on the y-axis of line in author's Fig. 3

x_1 = Maximum per cent pressure rise, or intercept on the x-axis of line in author's Fig. 3

E = Volumetric efficiency

y = Ordinate of author's line for "atmospheric volumetric efficiency"

Then

$$y = \frac{EP_4}{P_3}$$

$$x = 1 - \frac{P_4}{P_3}$$

At the end of the compression stroke, the volume C is occupied by air at P_3. During inlet stroke, this first expands to P_4 before any

new air is admitted. We may assume that this expansion is isothermal, due to the fact that there is a small body of air completely surrounded by a metal wall. Hence, the volume of expanded air in the clearance is $\frac{CP_3}{P_4}$. Hence, due to the effect of clearance only, volumetric efficiency is $1 - \frac{CP_3}{P_4}$.

If for other reasons we have volumetric efficiency losses D, we may then assume as a final semi-rational expression for volumetric efficiency,

$$E = D\left(1 - \frac{CP_3}{P_4}\right)$$

We take as a matter of definition $y = \frac{EP_1}{P_3E}$.

Hence $y = D\left(\frac{P_1}{P_3 - C}\right) = D\left(\frac{P_1}{P_3} - 1 - C + 1\right)$. Hence finally $y = D(1 - C - x)$. This theory gives y as a linear function of x as is given by the author's Fig. 3.

For $P_4 = P_3$, $x = 0$ and the ordinate becomes $y_1 = D(1 - C)$

For $\frac{P_1}{P_3} = C$, the abscissa becomes $x_1 = 1 - C$ and y becomes 0.

Hence the y intercept of the author's line is the product of the x intercept and the factor D.

If these laws can be shown to hold for any reciprocating compressor, as seems quite possible, all of the advantages the author cites of being able to approximate the volumetric efficiency curve may be accomplished for any compressor.

As the author points out, the tests he gives only show the action of a dry vacuum pump working as a reciprocating air compressor. The operation on an actual condenser with a mixture of air and saturated water vapor is quite a different proposition.

It would seem desirable for the author to give the actual formulae for the computation for the air handled for this case, as well as method of testing under the actual circumstances of the case, if any has been developed.

PAUL DISERENS. The paper points out the ease with which a volumetric test of a vacuum pump or compressor may be made, and emphasizes the high degree of accuracy to be obtained by the use of an orifice. It seems appropriate, however, to call attention to the fact that the vacuum which a pump will produce under a

closed suction is a good measure of its volumetric efficiency throughout its entire working range, and it will be interesting to note the degree of coincidence with which results thus calculated correspond with actual test results, such as those of the author.

Assuming that the volumetric loss is proportional to the number of compressions and remembering that the volumetric loss with suction closed exactly equals the displacement of the pump, the volumetric efficiency of a vacuum pump at any working vacuum is

$$E = 1 - \frac{a}{v}$$

where a = absolute pressure, in. hg., on closed suction
v = absolute pressure, in. hg., on working vacuum.
Converting this into terms of atmospheric volumetric efficiency as defined by the author,

$$E_a = \frac{1}{b}(v - a)$$

where b is the barometric pressure in in. of hg.

Fig. 5 gives a graphical comparison of the measured volumetric efficiency of a Worthington Feather valve vacuum pump for oil-field service, and the results calculated from the approximate formula based on vacuum developed under closed suction. The air measurements were made as described by the author except that a thin plate orifice was used and Durley's coefficients applied.

It should be noted that the two curves come so near together that for purposes of design and as a means of selecting a pump for any particular service, the approximate equation which is quoted above can be safely used, provided of course the usual factor of safety is applied.

A. H. Blaisdell. The author deals with the measurement of dry gases and not with moisture-laden gases such as would be handled by a dry-vacuum pump under operating conditions. While the writer has no particular criticism to make relative to the method of measurement cited by the author, it does seem rather misleading to run tests upon vacuum pumps, using dry air rather than a mixture of air and vapor, and then consider the results of such tests as indicating the actual capacity of the pumps.

Wm. G. Christy. The author states that the straight length of the nozzle beyond the curved mouth should not be less than one-quarter of the throat diameter, and may probably be as long

as the diameter without affecting the coefficient. In discussing this subject with users of such nozzles, the opinion seemed to be that this portion should be as short as possible.

H. Boyd Brydon. If dry vacuum pumps are to be tested, it should be done under the conditions under which they are to operate, with moist rather than dry air.

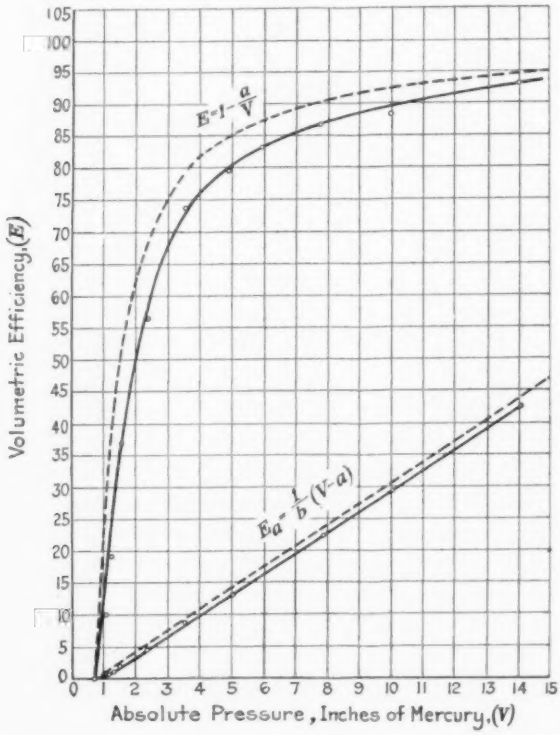


Fig. 5. Comparison of Measured and Calculated Volumetic Efficiency.

The Author: Reply to discussion by Dr. Sanford A. Moss.

The writer is much gratified by the words of appreciation expressed by this well known and capable experimenter. His remarks in regard to the application of the diagonal atmospheric volumetric efficiency line to an air compressor are much to the point because the writer has found this to be the actual fact. Plotting of many volumetric efficiency tests of single-stage air compressors, against variable pressure, shows a line similar to that observed for the

vacuum pump; but usually a little more curved — convex downward. The abscissae may be gage pressure, absolute pressure, or ratio of compression, and the ordinates would be volumetric efficiency expressed in air at intake conditions.

At first sight it would seem as if, with the compressor, the *diagonal* line plotted against volumetric efficiency in air at intake conditions would correspond with the *curved* line showing intake volumetric efficiency of the vacuum pump; but it will be observed on second thought that this is not the case. In the case of the diagonal line for the vacuum pump the air is measured at atmospheric pressure and intake temperature and therefore always at the same density. In this way the diagonal line shows air delivery in cubic feet of the same kind, irrespective of what the intake pressure or vacuum may be. In the case of the compressor, also using volumetric efficiency in air at atmospheric pressure and intake temperature, we have the same unit of measure, or air at constant density, and yet in the case of the compressor this automatically becomes air at intake conditions. Thus it will be seen that the diagonal "atmospheric volumetric efficiency" line of the vacuum pump should correspond to the volumetric efficiency of the compressor at its intake conditions.

Reply to discussion by Mr. Diserens.

His statements agree very closely with what the writer has observed, although he has found that in a compressor the volumetric loss plotted against ratio of compression is not exactly a straight line. Every design of compressor probably has its own characteristic line and there is a strong probability that the actual line depends upon the actual total cylinder clearance. There are so many factors which enter into the problem in an actual air compressor, depending upon jacket cooling, piston and valve leakage and the heating effect of the piston, that it is very difficult to separate these different factors. The actual ratio of volumetric loss to compression ratio appears to be not very far from a straight line, however.

Reply to Mr. Christy.

The question of the length of the straight part of the nozzle has been considered by many experimenters and it has been found that if the straight portion is too short there will not be a true parallel flow from the nozzle. This matter is being discussed by the various Air Compressor Test Code Revision Committees of the A.S.M.E. and it is the opinion that if the straight portion is about one quarter of the diameter, the best results will be obtained. Adding a little to this length would not appreciably increase the friction,

but cutting it down very much runs us into the danger of losing the parallel flow which is desired.

Reply to Messrs. Brydon and Blaisdell.

The author's paper is of course more applicable to the handling of dry or nearly dry gases than it is to a very wet mixture of steam and air. Some tests made by Mr. Paul Bancel of the Condenser Department of the Ingersoll Rand Company show that there is an appreciable loss of volumetric efficiency when vapor is present, this loss depending upon the amount of vapor. No doubt this result is produced by the re-evaporation of the steam in the clearance spaces. Mr. Bancel admitted steam to known quantities of air handled by a vacuum pump and tested by the author's method and he found the following volume losses with different ratios of partial air pressure to total pressure of mixture, the vacuum being 28 in.

Ratio Partial Air Pressure to Total Pressure of Mixture	Decrease in Capacity, per cent
1.00	0.00
0.50	8.00
0.40	14.3
0.30	26.0
0.25	34.0

This table shows that as the mixture becomes richer in steam, the capacity loss becomes greater, but it will be observed that the decrease is not very rapid as the quantity of steam first begins to increase.

With the above facts at hand it is advisable to bear in mind that there will be a decrease of capacity with the addition of steam; but knowing the dry-air capacity of the pump, which is easily obtained by the method described by the writer, a suitable allowance can be made to take care of the steam to be handled. As the quantities of air to be handled in a condensing plant must usually be figured on a liberal basis because of the many uncertainties involved, it is surely an advantage to have at hand at least the dry-air capacity of a given pump. This gives a measure of its best performance and thus reduces one important variable. For the handling of gases or air, moisture may be absent or greatly reduced. There are many commercial uses for dry-vacuum pumps; in the textile field and in the handling of paper, thin sheet metal and other light material which must be lifted and held during stamping, crimping and other operations, where there is no steam in the mixture and this effect need not therefore be considered at all.

No. 1791

REPORT UPON EFFICIENCY TESTS OF A 30,000-KW. STEAM TURBINE

By Herbert B. Reynolds, New York, N. Y.
Associate-Member

During 1920 the installation of three 30,000-kw. General Electric turbines was completed at the 59th Street power station of the Interborough Rapid Transit Company, New York City, complete tests upon one of which form the subject of the present paper.

These turbines are of the straight Curtiss impulse type, having 20 pressure stages, each pressure stage consisting of one velocity stage. The normal steam pressure at the throttle is 225 lb. per sq. in., abs. the steam being superheated 150 deg.fahr., and exhausted into a vacuum of 29 in. Hg. The speed is 1500 r.p.m. In addition to the primary steam inlet a secondary valve is provided which opens after the load reaches 24,000 kw. and which enables the turbine to carry a load of 35,000 kw.

The turbine tested was installed on a new reinforced concrete foundation. As no expansion joint was provided between the turbine and condenser, novel spring supports for the condenser were provided which are described in the paper.

The lowest water rate obtained during the tests while operating under normal conditions was 11.03 lb. per kw-hr., while the highest Rankine-cycle and thermal efficiencies obtained were 75.5 per cent and 25 per cent, respectively. Results of tests on the condensers and auxiliaries are also given in the form of tables.

The results obtained in these tests of a large power-station unit form a welcome addition to the data of tests of turbines of the same capacity installed in 1915 at the 74th Street power station of the company, reported in a paper presented before the Society in 1916, by H. G. Stott and W. S. Finlay, Jr.

IN order to provide additional power capacity for the new subways constructed in New York City during the period from 1913 to 1921, and operated by the Interborough Rapid Transit Company, additional turbine units were installed in both the 59th Street and 74th Street Power Stations. The three 30,000-kw. Westinghouse cross-compound turbines which were completed in 1915 were among the new units installed at the 74th Street Power Station. The results of the tests upon these three turbines were presented before the Society by H. G. Stott and W. S. Finlay, Jr., in 1916.[1]

[1] Trans. Am. Soc. M. E., vol. 38, page 655.

Presented at the Spring Meeting, Chicago, Ill., May 23 to 26, 1921, of The American Society of Mechanical Engineers, New York.

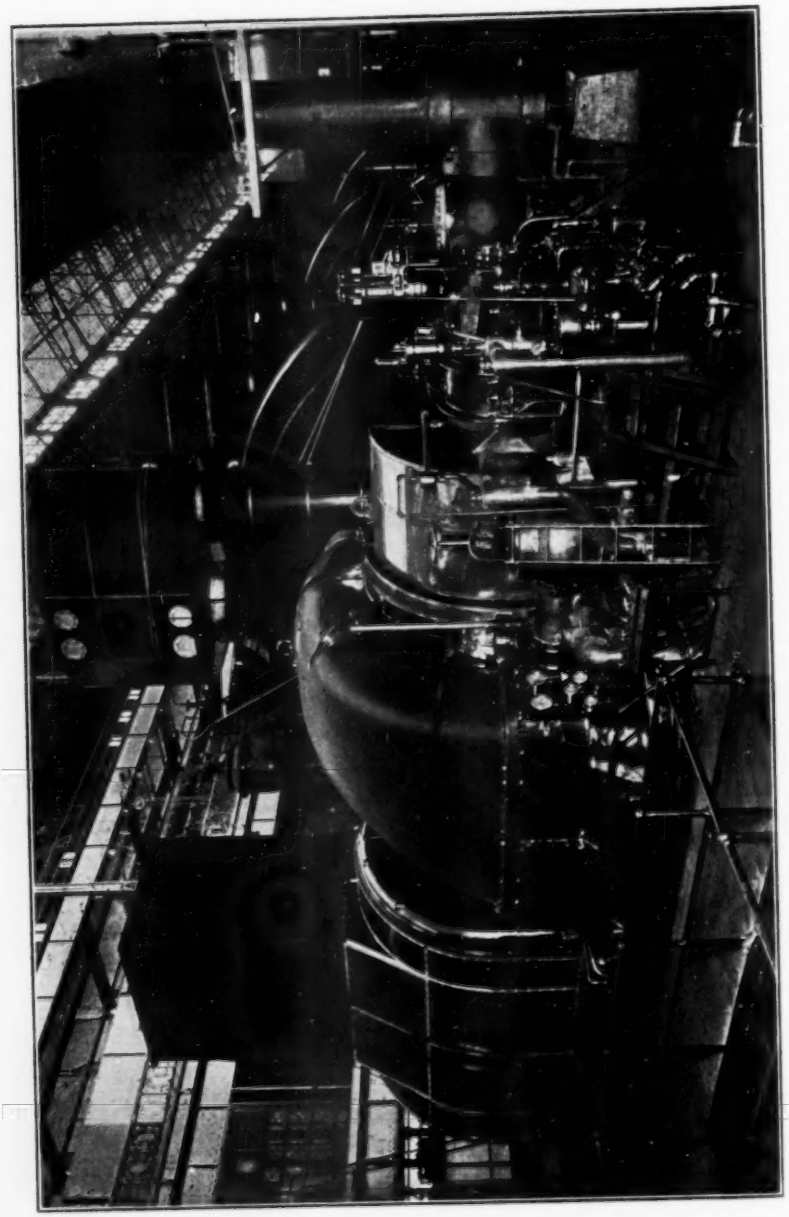


FIG. 1 ONE OF THE THREE 30,000-KW. TURBINES INSTALLED AT THE 59TH ST. POWER STATION OF THE INTERBOROUGH RAPID TRANSIT CO.

2 At the 59th Street Power Station, the installation of three 30,000-kw. General Electric turbines has been completed within the past year. Very complete tests have been conducted upon these units, the results of which are included in this paper.

3 Reviewing briefly the history of the developments at the 59th Street Power Station, it will be recalled that the original engine-room equipment consisted of nine 7500-kw. maximum-capacity Manhattan-type Allis-Chalmers double angle compound engine units, and three Westinghouse 1250-kw. turbines, the latter driving 60-cycle generators which supplied 60-cycle current for subway lighting. The use of this current for lighting was subsequently discontinued and in its place 25-cycle current is now used which is taken from the main units. During 1909 and 1910, five low-pressure 7500-kw. maximum-capacity General Electric turbine units were added, taking exhaust steam from five of the engines at atmospheric pressure. A description of these units and a full report of the tests which were conducted on them are given in a paper presented by H. G. Stott and R. J. S. Pigott before the Society in 1910.[1] The units most recently installed in the 59th Street Power Station are the three 30,000 kw. General Electric turbines mentioned above.

THE TURBINE UNITS

4 Two of the 30,000-kw. units, Nos. 7 and 8, were installed in the space formerly occupied by the three lighting units mentioned in the preceding paragraph, while the third turbine, unit No. 12, see Fig. 1, was installed at the western end of the station. The great concentration of power possible with modern turbines is strikingly shown by the space they require as compared with that for reciprocating engines. The maximum capacity of the engines visible in the figure is but 26,250 kw. while that of the turbine in the foreground is 35,000 kw.

5 The general arrangement of unit No. 12, upon which the tests were conducted, is shown in Fig. 2. The turbines are of the straight Curtis impulse type, having twenty pressure stages, each pressure stage consisting of one velocity stage. A cross-section of the turbine is shown in Fig. 3, and a general view of the unit showing both the turbine and condenser is given in Fig. 4. Fig. 1 is a close view of the governor end of the turbine and shows the auxiliary oil pump, etc.

[1] Trans. Am. Soc., M. E., vol. 32, page 69

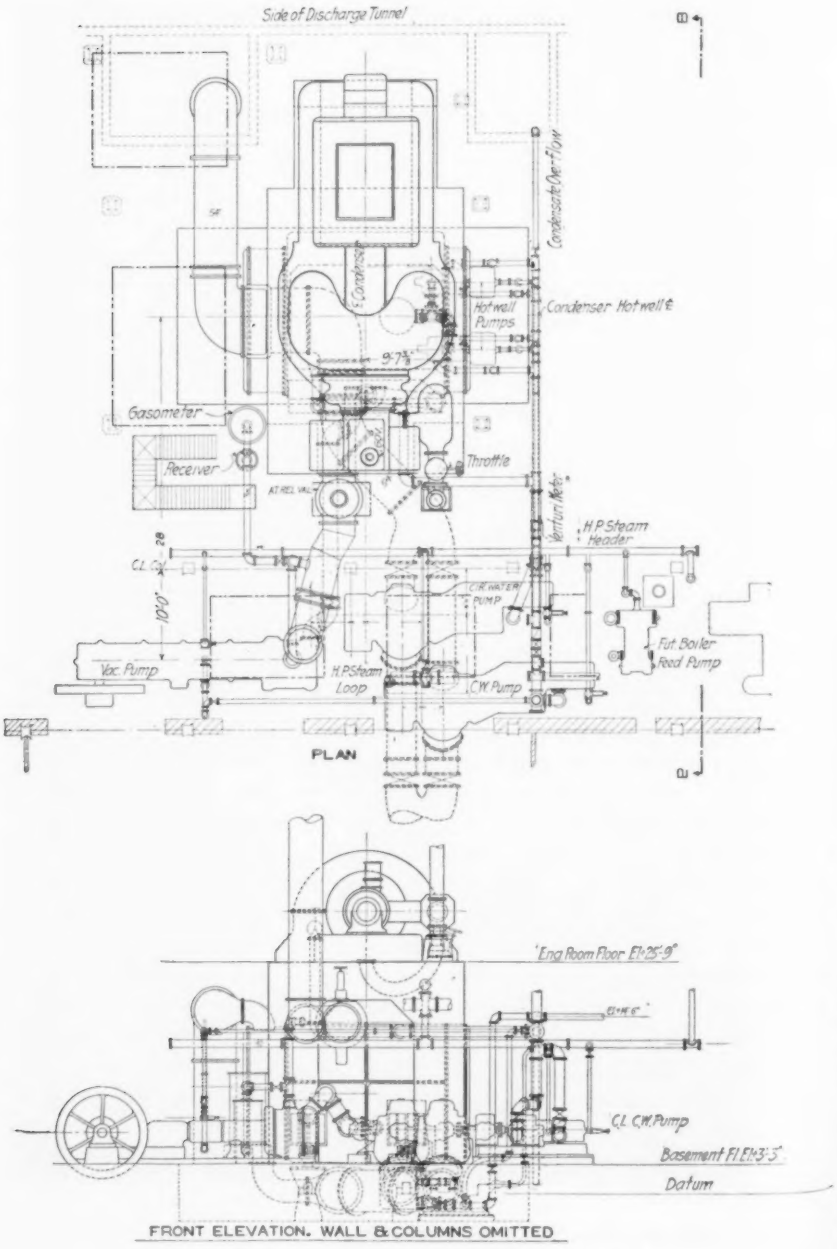
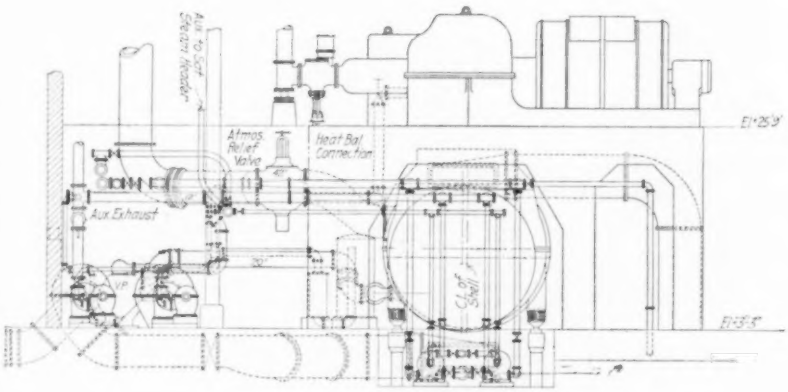


FIG. 2 GENERAL ARRANGEMENT OF UNIT NO. 2

(See also Fig. 2A)

6 The normal steam pressure at the throttle is 225 lb. per sq. in. abs., with a superheat of 150 deg. fahr. exhausting into a vacuum of 29 in. Hg referred to a 30-in. barometer at 58.1 deg. fahr. The speed is 1500 r.p.m. In addition to the primary steam inlet, a secondary valve is provided which opens after the load reaches 24,000 kw. and which enables the turbine to carry a load of 35,000 kw. The generator is designed so that this load may be carried continuously.

7 Water-sealed glands are used which obtain their water supply from the condensate discharge. A self-contained lubricating system is provided. The oil is circulated through the coolers and bearings



ELEVATION B-B

FIG. 2A ELEVATION B–B IN FIG. 2

by means of a pump driven from the turbine shaft. For starting up and emergency purposes a separate pump is provided which is driven by a small steam turbine, the speed of which is automatically controlled by the oil pressure. In addition to the cooling effect of the oil, the bearings are further cooled by the circulation of condensate through water jackets.

8 As all auxiliaries in the station are steam-driven, a connection has been provided in the turbine through which any excess auxiliary steam may be injected. This connection is at the 16th stage of the turbine, as shown in Fig. 3.

9 The generators are three-phase, star-connected, generating 25-cycle current at 11,000 volts. Excitation current is furnished at

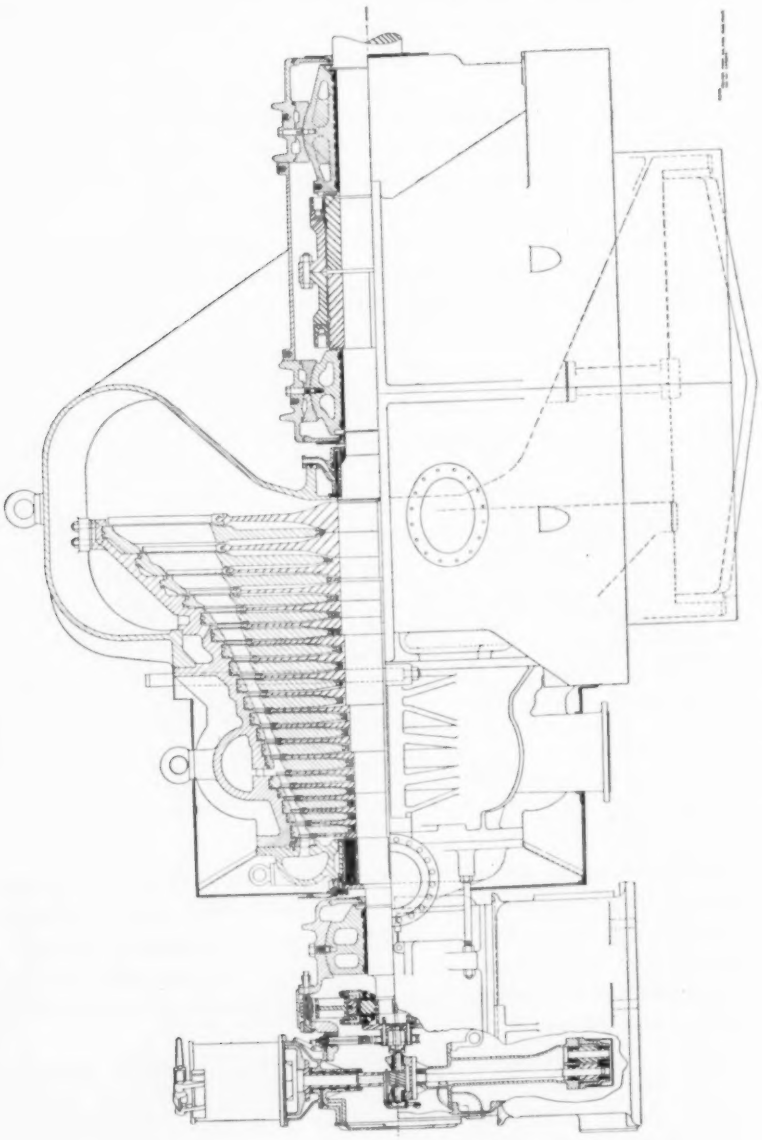


FIG. 3 CROSS-SECTION OF TURBINE SHOWN IN FIG. 1

250 volts. The generators are cooled by a circulation of air maintained by a fan which forms an integral part of the generator. The air is drawn from the turbine-room basement and discharged from the top of the generator into the turbine room through a short stack which may be seen in Figs. 1 and 4.

CONDENSERS AND AUXILIARIES

10 The condensing equipment for each unit consists of one single-shell two-pass Worthington condenser, two Worthington centrifugal circulating pumps each driven through reduction gears by Kerr turbines, two Worthington centrifugal condensate pumps each driven by a General Electric turbine, and one Laidlaw-Dunn-Gordon dry-vacuum pump. The general arrangement of the condensing equipment is shown in Fig. 2.

11 Each condenser contains 50,000 sq. ft. of tube surface made up of 10,760 tubes 18 ft. long, 1 in. in outside diameter and of No.18 B. W. G. thickness. Fig. 4 gives a view of unit No. 12 including the condenser. The condenser is of the two-pass type, the water entering the bottom and passing out from the top. The discharge circulating-water pipe may be seen in Fig. 4. As the condensers are mounted on springs, rubber expansion joints are inserted in the circulating-water lines.

12 Each circulating-water pump is capable of delivering 30,000 gal. of water per min. against a total head of 37 ft. The turbines which drive these pumps operate at 3950 r.p.m. This speed is reduced to 395 r. p. m. through Kerr reduction gears. The impellers are made of bronze and are of the enclosed or shrouded design.

13 Each condensate pump is capable of delivering 950 gal. per min. against a discharge head of 60 ft. The turbine and pump both operate at a speed of 1500 r.p.m.

14 The dry-vacuum pumps are of the single rolling-mill frame two-stage rotative type, with poppet-valve steam cylinder and two-stage water-jacketed vacuum cylinder.

FOUNDATIONS

15 Two of the units, Nos. 7 and 8, were installed on the foundations which formerly carried the three 1250-kw. 60-cycle lighting units mentioned in Par. 3. However, it was necessary to alter these foundations to some extent in order to receive the new turbines. The

third turbine, unit No. 12, was installed on new reinforced concrete foundations.

16 As no expansion joint was provided between the turbine



FIG. 4 VIEW OF TURBINE AND CONDENSER, UNIT NO. 12

and condenser, it was necessary to mount the condenser on spring supports so as to take care of the expansion and contraction. These spring supports are shown in Fig. 5.

17 In order to facilitate the setting of these springs and also to provide a means for detecting and adjusting for any fatigue which might occur in them, hydraulic jacks were incorporated in the condenser supports as shown in Fig. 5. As the procedure followed in setting these springs may be of general interest, a brief description will be given. After the erection of the condenser and circulating-water pipe had been completed, with the exception of making the

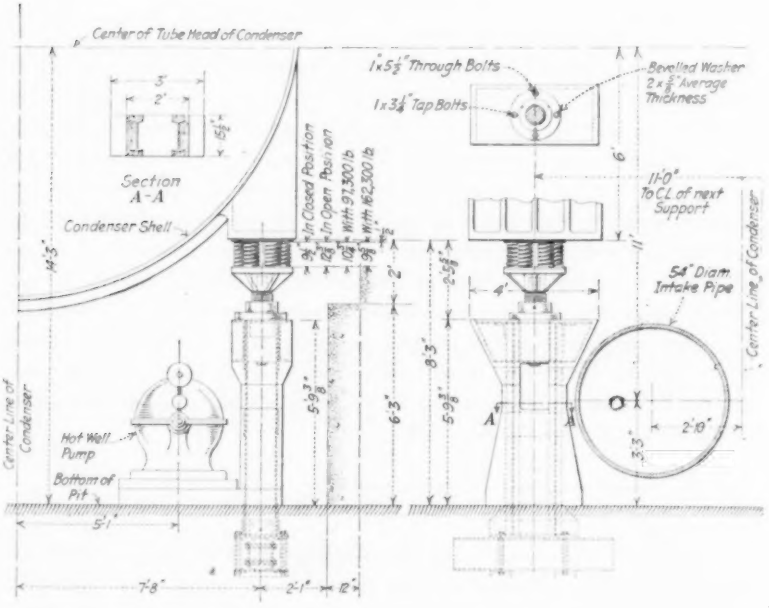


FIG. 5 SPRING SUPPORTS FOR CONDENSER

joint between the condenser and turbine, the condenser was raised while empty, by means of the jacks, leaving $\frac{3}{8}$ in. clearance between the face of the turbine outlet and the face of the condenser inlet. The load on each of the four supports was then determined by noting the oil pressure in the jacks. It was decided that, with the condenser empty and cold, the downward pull on the turbine should not be less than approximately 17 tons. The distance that the joint between turbine and condenser would have to be pulled in order to give this load was estimated from the modulus of elasticity of the turbine and condenser metal. The condenser was then raised to within the

predetermined distance of the turbine outlet, which was found to be 0.231 in., after which the lock nuts on the jacks were screwed home and the condenser bolted to the turbine. The load on the springs was then determined with the condenser still empty by noting the pressure required to just raise the lock nuts. Every few months the load carried by the springs will be determined in this manner and compared with the load which existed when the condenser was first bolted to the turbine. Any fatigue which may develop in the springs will be compensated for by screwing the lock nuts down.

18 It was found that the minimum condenser load carried by the turbine with the condenser shell empty was approximately 17 tons, which checked with the established minimum. As the water required to fill the condenser amounts to approximately 60 tons, the load on the turbine increases to 77 tons when the circulating-water pumps are started. However, as the result of a slight expansion of the condenser which takes place when it is warmed up, the springs are compressed and a small part of this load is transferred to the supports, reducing the load on the turbine to approximately 70 tons under operating conditions. Immediately after shutting down and while the condenser is still warm but drained, the load on the turbine is reduced to 10 tons. It will thus be seen that the condenser load on the turbine varies from 10 to 77 tons. The total weight of the condenser varies from 180 to 240 tons.

TURBINE TESTS

19 The equipment used for conducting the turbine tests consisted of two large water-weighing scales for measuring the steam consumption, three single-phase rotating standard watt-hour meters for measuring the output, and all the necessary thermometers, gages and mercury columns for determining temperatures, pressures and vacua as recorded in this report.

20 The water-weighing scales had a capacity of about 25,000 lb. each. These scales were very carefully calibrated several times throughout the tests by standard test weights and the total steam consumption corrected accordingly. The rotating standard watt-hour meters which were used for measuring the output were calibrated before and after the tests by the Electrical Testing Laboratories, New York City. It is believed that both the input and output were measured within an accuracy of 0.25 per cent. All thermometers and gages were also calibrated in the usual manner. The specific

gravity of the mercury used in the vacuum columns was determined and corrections made accordingly. The readings obtained from the mercury columns were further corrected for meniscus, temperature, and barometer reading. The barometer reading was obtained from the local U. S. Weather Bureau.

21 Most of the tests were of three hours' duration. With the exception of a few special tests, the turbine was operated under

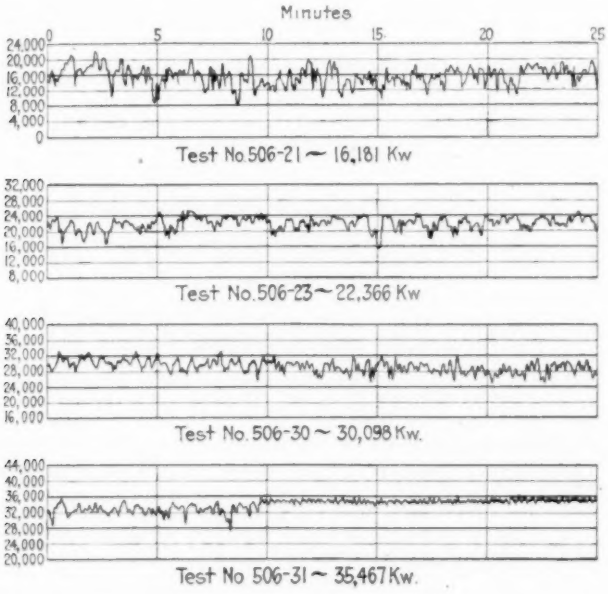


FIG. 6 REPRESENTATIVE GRAPHIC LOAD CHARTS SHOWING EXTENT OF SWINGS

normal conditions in so far as type of load was concerned. The load was controlled from the switchboard through the remote governor-control system provided for that purpose. This method of controlling the load subjected the turbine to the full swings of the railroad load. Fig. 6 shows representative graphic load charts taken during the tests, from which may be seen the extent of the swings under which the turbine was operating. The evenness of the load shown during part of test No. 31 is due to the fact that the governor was so adjusted as to limit the load to a little over 35,000 kw., which prevented the unit from taking any of the swings.

TABLE 1 SUMMARY OF TURBINE TEST RESULTS

Test No.	Load Average, net kw.	Duration hr.	Steam Pressures, lb. per sq. in. abs. Throttle	Turbine head	Overload, 7th stage	Heat-balance connection, 16th stage	Exhaust vacuum in. Hg[1]	Steam temp. at throttle, deg. fahr.	Superheat, deg. fahr.	Water, actual total, lb.	Water, corr. total,[2] lb.	Output, gross total, kw-hr.	Excitation,[3] kw-hr.	Output, net total, kw-hr.	Water rate corr.,[2] lb. per kw-hr.	Rankine cycle efficiency,[2] per cent	Thermal efficiency,[2] per cent
36	13,985	3	237	138	82	8.13	29.00	551	155	489,140	492,786	42,118	162	41,956	11.75	70.80	23.46
44	14,146	3	231	142	79	7.60	28.94	537	143	501,560	495,858	42,608	171	42,437	11.68	71.23	23.60
20	14,161	3	231	134	80	7.88	28.99	550	156	498,125	501,233	42,643	161	42,482	11.80	70.51	23.36
33	14,175	3	236	136	81	8.08	28.92	559	163	492,215	495,984	42,686	161	42,525	11.66	71.36	23.64
32	16,165	3	233	150	87	8.78	28.92	548	153	557,730	556,228	48,664	168	48,496	11.47	72.54	24.04
21	16,181	3	229	149	89	8.81	28.97	548	155	555,710	557,438	48,712	169	48,543	11.48	72.47	24.02
10	17,889	3	235	165	95	10.33	28.90	498	103	640,452	607,640	53,845	178	53,667	11.32	73.50	24.38
25	18,243	3	227	166	99	10.25	28.96	546	153	617,170	617,047	54,905	176	54,729	11.28	73.76	24.44
24	20,142	3	229	182	108	11.61	28.96	524	131	688,785	674,156	60,611	185	60,426	11.16	74.55	24.70
11	20,150	3	235	188	114	11.63	28.93	501	105	706,687	673,677	60,636	186	60,450	11.14	74.69	24.74
12	22,065	3	224	196	119	12.87	28.87	513	122	764,761	735,842	66,389	194	66,195	11.12	74.82	24.80
35	22,217	3	232	194	114	12.38	29.00	552	157	731,290	737,485	66,848	196	66,652	11.07	75.16	24.91
23	22,366	3	228	198	117	12.22	28.95	520	127	758,280	738,560	67,293	196	67,097	11.01	75.57	25.04
13	23,945	3	233	207	124	13.75	28.96	518	123	814,978	792,241	72,038	201	71,837	11.03	75.43	25.00
22	24,151	3	227	206	124	13.48	28.99	523	130	815,910	799,598	72,657	203	72,454	11.04	75.36	24.97
42	24,159	3	228	212	119	13.40	29.01	514	121	822,875	800,735	72,681	203	72,478	11.05	75.29	24.95
34	26,206	3	231	216	132	14.92	28.92	540	146	873,350	865,389	78,830	213	78,617	11.01	75.57	25.04
19	26,357	3	224	207	141	15.28	29.01	519	127	898,485	878,714	79,284	213	79,071	11.12	74.82	24.79
14	26,400	3	228	216	139	15.31	28.91	526	133	905,215	884,344	79,425	224	79,201	11.17	74.49	24.68
15	28,244	3	228	216	151	17.02	28.95	523	130	968,485	946,341	84,954	223	84,731	11.17	74.49	24.68
18	28,249	3	225	211	159	18.71	28.84	518	126	979,565	944,796	85,269	221	85,048	11.11	74.89	24.81
30	30,098	3	224	212	161	17.93	29.00	527	136	1,028,595	1,014,193	90,523	228	90,295	11.23	74.09	24.55
43	30,289	3	225	222	161	17.20	28.97	516	124	1,048,920	1,020,053	91,100	233	90,867	11.23	74.09	24.55
17	30,470	3	223	212	165	18.45	29.02	514	123	1,055,025	1,027,989	91,641	231	91,410	11.25	73.96	24.50
41	33,301	3	225	222	177	20.16	28.84	502	110	1,177,895	1,119,672	100,148	245	99,903	11.21	74.22	24.59
40	34,448	3	224	221	186	20.67	28.96	498	106	1,221,695	1,166,630	103,600	255	103,345	11.29	73.63	24.40
37	35,254	3	224	217	191	21.84	29.03	497	105	1,253,160	1,200,517	106,023	260	105,763	11.35	73.30	24.29
31	35,467	3	224	216	193	21.53	28.99	521	130	1,224,235	1,199,055	106,660	258	106,402	11.27	73.82	24.46
26[4]	12,991	2	235	128	75	8.35	28.98	538	142	311,700	309,257	26,085	103	25,982	11.90	69.92	23.17
29[4]	20,073	2	217	178	105	11.42	29.03	528	139	452,960	448,298	42,270	125	40,145	11.17	74.49	24.68
28[4]	25,733	2	226	209	136	13.23	28.97	519	127	587,150	572,718	51,605	140	51,465	11.13	74.75	24.77
16[5]	25,196	3	228	225	133	14.62	28.94	533	140	867,385	855,156	78,798	210	78,588	10.88	76.47	25.34
27[4]	31,568	2	223	212	171	12.13	28.99	521	130	727,290	712,192	63,292	156	63,136	11.28	73.76	24.44
38[6]	No load	0.5	234	23	17		28.15	499	104	8,925			22				
39[7]	No load	0.5	234	23	17		27.93	502	107	8,045							

For notes to table see following page.

NOTES TO TABLE 1

[1] Referred to 30-in. barometer at 58.1 deg. fahr.

[2] As corrected to 225 lb. per sq. in. abs. pressure, 150 deg. fahr. superheat, 29 in. Hg vacuum ref. to 30-in. barometer at 58.1 deg. fahr.

[3] Including rheostat loss.

[4] Tests Nos. 26, 27, 28 and 29 made with operating governor adjusted to limit maximum loads to values shown in 2d column.

[5] Test No. 16 made with secondary valve out of service.

[6] Test No. 38 made without load on generator, with generator field current adjusted to give normal voltage on open circuit.

[7] Test No. 39 made without load on generator, generator field not excited.

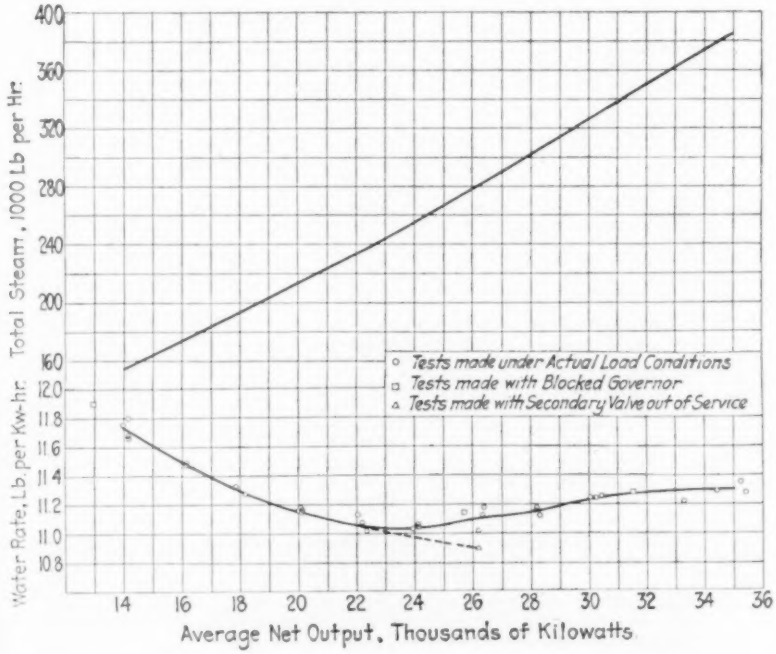


FIG. 7 WATER-RATE CURVE AND WILLANS LINE FOR 30,000-KW. TURBINE

Steam pressure, 225 lbs. abs.; superheat, 150 deg. fahr.; vacuum, 29 in. Hg, referred to a 30-in. barometer; water-rate factor, 0.985. Curve based on tests Nos. 10 to 44, inclusive (excluding special tests with blocked governor and test with secondary valve out of service).

22 Table 1 gives the numerical results of the turbine tests, while the performance is shown graphically in Figs. 7, 8, and 9. Fig. 7 gives the total steam consumption and water rate of the unit, while Fig. 8 gives the thermal and Rankine-cycle efficiencies. From these curves it will be seen that the lowest water rate obtained while

operating under normal conditions was 11.03 lb. per kw-hr., while the highest Rankine and thermal efficiencies obtained were 75.5 per cent and 25.0 per cent, respectively. As stated above, all tests with the exception of seven were conducted under operating conditions.

23 In order to determine the effect of the swinging load on the steam consumption, four tests were conducted with the governor blocked so that the unit operated under a steady load. These tests

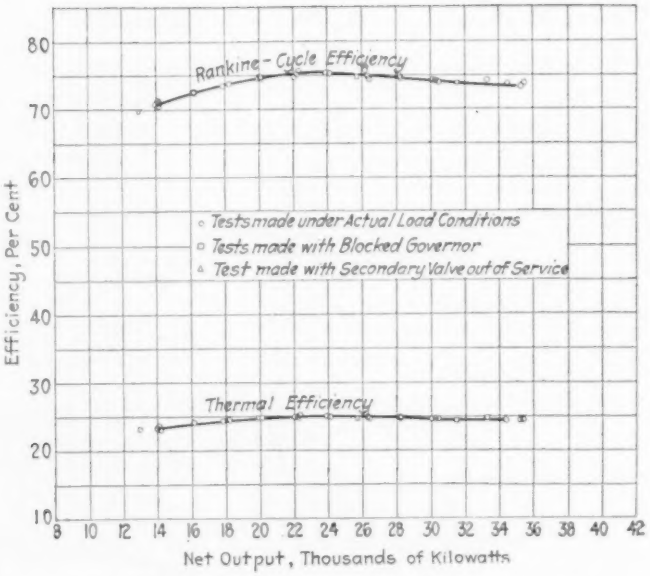


FIG. 8 RANKINE-CYCLE AND THERMAL EFFICIENCY CURVES FOR 30,000-KW. TURBINE

Steam pressure, 225 lb. abs.; superheat, 150 deg. fahr.; vacuum, 29 in. Hg, referred to 30-in. barometer. Curves based on tests Nos. 10 to 44, inclusive (excluding special tests with blocked governor and test with secondary valve out of service).

are shown in Figs. 7 and 8 by the small squares. It will be seen that there is no improvement in the efficiency under the steady-load conditions at the loads selected for these tests. When the turbine is operating under normal conditions and carrying loads from 22,000 kw. to 25,000 kw. the secondary valve is continually opening and closing, which no doubt reduces the economy somewhat. If steady-load tests had been conducted within this range the points would have probably fallen on the dotted curve. The test shown by the small triangle in Figs. 7 and 8 was run with the secondary or overload

valve closed and out of commission, while the load was adjusted so that the primary valve was wide open. As is to be expected, the lowest water rate was obtained while operating in this manner. Since the completion of these tests the governor cams have been modified so that the opening of the secondary valve is delayed. This may improve the economy to some extent near and at the best load point; however, no tests have been conducted to determine just how much improvement has been accomplished.

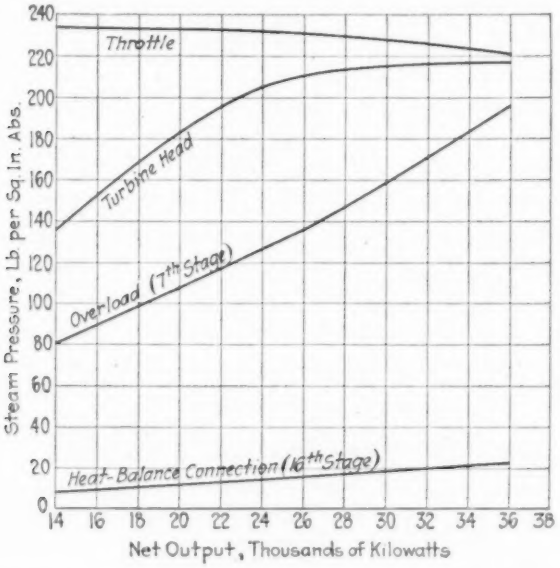


FIG. 9 VARIATION OF STEAM PRESSURE WITH THE LOAD AT VARIOUS POINTS THROUGHOUT THE TURBINE

(Based on tests Nos. 10 to 44, inclusive).

24 Test No. 38 was run without any load on the generator, but with the field excited to give normal voltage on open circuit. The actual steam consumption per hour during this test amounted to 17,850 lb. Another no-load test was conducted, but without any excitation, during which the actual steam consumption amounted to 16,090 lb. per hour. No attempt was made to correct the steam consumption during these two no-load tests, due to the vacuum being low and outside the range of the correcting curves.

25 Fig. 9 shows the variation of steam pressure with the load at various points throughout the turbine. In order to correct the

steam consumption to the standard conditions, the curves given in Fig. 10 were used.

WATER-RATE FACTOR

26 In order to make an absolute and numerical comparison of the flatness of water-rate curves for different turbines, an expression

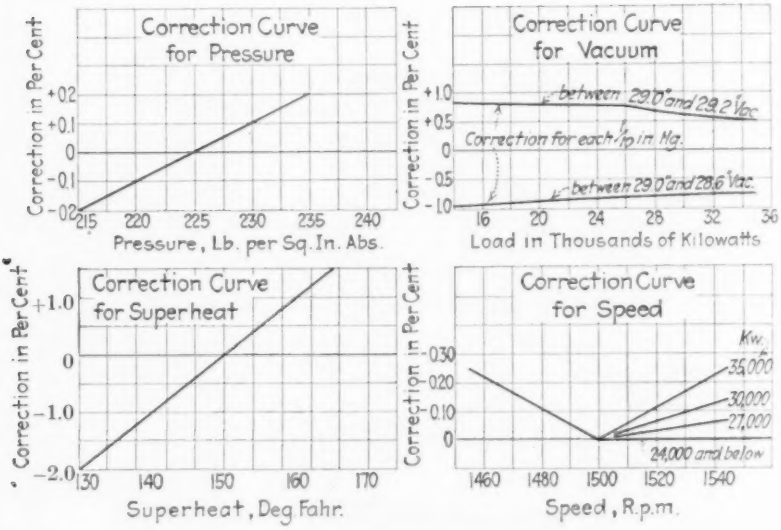


FIG. 10 CURVES FOR CORRECTING STEAM CONSUMPTION TO THE STANDARD CONDITIONS

has been used which is termed the "water-rate factor." Expressed mathematically,

$$F = \frac{MW}{C}$$

in which

F = water-rate factor
M = minimum water rate obtained
W = mean load between one-half rated load and maximum load, which is the usual operating range.
C = Average total steam consumption per hour covering the range between one-half rated load and maximum load.

27 Using the curves given in Fig. 7, substitutions may be made as follows:

$$F = \frac{11.03 \times 25{,}000}{280{,}100} = 0.985$$

This factor could be expressed by using the ratio between the minimum water rate and the average water rate over the range selected. However, the factor obtained in this manner would not take into account the fact that it is more desirable to obtain flatness in water-

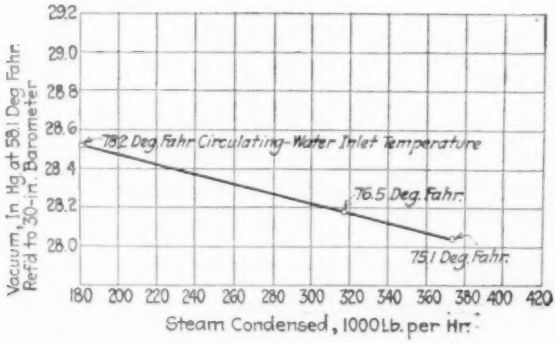


FIG. 11 CURVE SHOWING VACUUM OBTAINED DURING CONDENSER TEST

Based on tests Nos. 2, 3 and 4. Circulating-water inlet temperature varied from 75.1 deg. fahr. to 78.2 deg. fahr. Condenser surface, 50,000 sq. ft.

rate curves at the high loads than it is at the low loads. By using the total steam consumption as indicated in the expression given above, a fairer comparison is obtained.

CONDENSER AND AUXILIARY TESTS

28 Tests were conducted on the condensers and auxiliaries. A summary of the condenser and dry-vacuum-pump tests, which were run simultaneously, is given in Table 2, while the relation between load and vacuum is shown graphically in Fig. 11. In examining the results of these tests and comparing them with the vacuum obtained during the turbine tests, it should be kept in mind that the temperature of the circulating water during the condenser tests, which were run during the summer, was very high, varying

from 75.1 deg. fahr. to 78.2 deg. fahr. The methods used in determining the quantities given in Table 2 need no description, with the possible exception of the measurement of the air discharged from the dry-vacuum pump and the steam consumption. The air was measured by a gasometer connected to the discharge of the pump.

TABLE 2 SUMMARY OF CONDENSER AND DRY-VACUUM-PUMP TESTS

Number of Test	2	3	4
CONDENSER TESTS			
Average load, lb. steam per hour	180,666	317,000	373,666
Duration of test, hours	3	3	3
Barometer, at 58.1 deg. fahr.	29.93	29.93	30.07
Superheat at turbine throttle, deg. fahr.	41	44	50
Steam pressure at turbine throttle, lb. per sq. in. abs.	230	225	225
Vacuum, in. Hg. at 58.1 deg. fahr. referred to 30-in. barometer	28.51	28.17	28.04
Temperature corresponding to vacuum, deg. fahr.	91.6	98.3	100.6
Temperature of circulating water in, deg. fahr.	78.2	76.5	75.1
Temperature of circulating water out, deg. fahr.	83.0	86.3	86.4
Rise in temperature of circulating water, deg. fahr.	4.8	9.8	11.3
Temperature of condensate water, deg. fahr.	87.0	91.3	91.5
Circulating water (two pumps running), gal. per min.	72,500	61,500	63,800
Heat transferred per hour, B.t.u.	174,144,000	301,277,000	360,662,000
Heat transferred per hour per sq. ft. surface, B.t.u.	3483	6025	7213
Mean temperature difference (log.), deg. fahr.	10.8	16.36	19.4
Heat transferred per sq. ft. surface per hour per deg mean temperature difference	322	368	372
DRY-VACUUM-PUMP TESTS			
Vacuum at pump, in. Hg. at 58.1 deg. fahr. referred to 30 in. barometer	28.58	28.55	28.56
Temperature of air and vapor entering pump, deg. fahr.	82	82	82
Temperature of air and vapor leaving pump, deg. fahr.	185	167	160
Temperature of jacket water entering pump, deg. fahr.	68	63	63
Temperature of jacket water leaving pump, deg. fahr.	89	73	71
Speed of pump, r.p.m.	48	47	48
Steam pressure at throttle, lb. per sq. in. abs.	209	207	208
Superheat at throttle, deg. fahr.	11	17	20
Steam pressure at exhaust, lb. per sq. in. abs.	15.8	15.9	15.6
Steam consumption, lb. per hr. corrected to 225 lb. abs. steam pres., 150 deg. fahr. superheat, 16 lb. abs. exhaust pressure	1088	1029	964
Air discharged, cu. ft. per min. at 60 deg. fahr.	9.94	9.47	6.50

The steam consumption was determined by condensing the exhaust steam in a small test condenser, after which it was weighed in the usual manner.

29 Table 3 gives the results of the tests on the circulating-water pump. It will be seen that two tests were conducted on these pumps, one being an economy test while the other was a maximum-capacity test. Only one of the two pumps was running in either

test. During the economy test the discharge valve was adjusted to give the normal operating head which exists when two pumps are running, while in the capacity test it was left wide open. The quan-

TABLE 3 SUMMARY OF CIRCULATING-WATER-PUMP TESTS

ECONOMY TEST[1]	
Duration of test, hours	2
Barometer, at 58.1 deg. fahr	29.78
Steam pressure at throttle, lb. per sq. in. abs	22
Superheat at throttle, deg. fahr	21
Steam pressure at exhaust, lb. per sq. in. abs	15.1
Speed of pump, r. p. m	386
Pump discharge, gal. per. min.	33,300
Total head on pump, ft	37.63
Steam consumption, lb. per hour corrected to 150 deg. fahr. superheat	10,860
CAPACITY TEST[2]	
Speed of pump, r. p. m	382
Pump discharge, gal. per min	51,800
Total head on pump, ft	20.60

[1] Test made with discharge valve adjusted to give approximately normal operating head.
[2] Test made with discharge valve wide open.

tity of water pumped was computed by the heat-balance method, which involved the measurement of the main-turbine condensate, its temperature, the temperature of the circulating water in and out,

TABLE 4 SUMMARY OF CONDENSATE-PUMP TEST

Duration of test, hours	1
Barometer, at 58.1 deg. fahr	29.79
Steam pressure at throttle, lb. per sq. in. abs	216
Superheat at throttle, deg. fahr	28
Steam pressure at exhaust, lb. per sq. in. abs	15.2
Speed of pump, r. p. m	1495
Pump discharge, lb. per hour	475,000
Suction head, in. Hg	26.2
Discharge head, ft	60.9
Total head on pump, ft	90.64
Steam consumption, lb. per hour corrected to 150 deg. fahr. superheat	1840

the steam conditions at the main-turbine throttle, load on main unit, etc. The steam consumption of the turbine driving the pump was determined by means of the small test condenser mentioned above. The results of the condensate pump test are given in Table 4.

DISCUSSION

W. S. FINLAY, JR. The author's report is a valuable contribution to the Society and a notable one of a series made by the Motive Power Department of the Interborough Rapid Transit Company and published in accordance with the policy initiated by the late H. G. Stott.

As, until recently, head of the department responsible for the general design of the installation, the writer naturally has a certain amount of personal pride in its success and particularly in having contributed to the paper to the extent of placing such an able and thorough investigator as Mr. Reynolds in charge of the tests reported therein.

The tests report details are self-explanatory and comment thereon would be superfluous.

The writer's chief interest is, however, in the details of the installation which, if merely superficially examined, might be subject to criticism, and which, therefore, in all fairness should be presented with a few brief explanatory notes.

Preliminary to discussing these, certain ruling considerations affecting the design must first be noted, viz, the high price of New York real estate and the necessity and desirability of so locating the turbines as to secure their steam supply with proper distribution from the existing boiler plant.

Space for installation was available at two points of the plant; one near the center of the engine room floor, where in the early years of the plant, as referred to in Par. 15, certain small lighting units had originally been installed whose use had been abandoned for some years as incidental to changes in the general lighting system; the other in unused foundation room at the west end of the building. Two machines were placed in the first-named space and one in the latter. The adaptation of these old foundations furnished a unique and interesting problem, it only being possible to appreciate the reconstruction details by inspection of the work in the field. The exhaust end of the turbine was carried on a bridge supported by the rebuilt concrete foundations and although in appearance it was seemingly extremely light, nevertheless in service it has proven to be unusually substantial and free from vibration.

In order to introduce what was felt to be the requisite number of auxiliaries, a "shoe-horn" policy had to be followed; hence,

for instance, the peculiar arrangement of the circulating water pumps shown in Fig. 2. The intake was by pipe from the plant's original intake tunnel. The capacity of such tunnel was figured as capable of carrying the installation to the extent of loads anticipated in the immediate future. The peculiarity of this installation of the pump lies in the overlap, the end bearing of one pump being carried on a bridge over the intake of the other, and the steam end of the second pump bridging the discharge of the first. Ample protection against distortion by differential expansion is introduced by rubber expansion joints. The discharge of the turbines was carried into a new discharge tunnel, built as part of the installation.

The building of this tunnel, which passes under the water end of the foundations of the plant's large reciprocating units, furnished a most interesting piece of construction work. The hot wells at the bottom of the barometric-tube condenser had to be removed in the course of this construction. They were substituted for by large temporary wooden wells, the engines actually being operated during the construction period, discharging into these temporary wells. This type of operation was only one incident in a number furnishing interesting operating problems during the construction period.

The author has referred in Par. 16 to the spring supports for the condensers. The writer's design of these supports was prompted by a chance remark made by H. G. Stott, shortly before his death and prompted by his curisoity as to the amount of fatigue developing in the springs supporting the condenser in the 74th Street plant. The writer personally was also somewhat curious to determine whether the ordinary method of bolting up the condenser to the turbine exhaust did not introduce most uneven strains upon the exhaust casting. Hence he developed the hydraulic jack and locknut idea. His curiosity was satisfied when it was discovered that the strains noted would have been most unevenly distributed had ordinary methods been applied. The writer is egotistical enough to consider this feature of the installation as somewhat unique and valuable.

Another feature which might call for criticism is the pit beneath the condenser. This was necessitated by the desire to avoid changing the level of the operating floor.

Again, critics might wonder why a stack was installed upon the air discharge of the generator. It would have been preferable to have disposed of the heated air in other fashion, in ways quite

common in recent installations, but space limitations and conditions largely prevented this. The stack directs both heated air and sound toward the roof and reduces objectionable conditions, which would otherwise be caused, to a minimum.

The installation of the steam piping incidental to the job might be well worth the description did space and opportunity permit. The writer can only say that the design, for which he can assume personal responsibility, carries in it every possible opportunity for unrestricted expansion and avoidance of local strains. Incidentally, these distributing mains, which have replaced the original system through about half the house, were installed under conditions which required fairly continuous use of the old mains right up to and after making the change. There was necessitated, therefore, the development of a system which could be installed with as little interference as possible with the old horizontal triple zig-zag system with which many are familiar.

The development of the system of boiler operation at high ratings, "tucking in" of an induced draft system, etc., permitting to carry the increased load with the original boiler also furnish interesting contributions to the history of the job.

The writer cannot possibly give expression to the difficulties of making this change, but it unquestionably reflected enormous credit upon the men immediately in charge of construction, Messrs. A. B. Williams, now with Stone and Webster, and W. F. Ryan, now with the Hope Engineering Company of Boston. Naturally, the care of the operating men to avoid mistakes and error incidental to the huge and complex changes which were daily taking place cannot be too highly praised and the writer believes that proper credit should be given to Willis Lawrence, the mechanical engineer in charge.

There were numerous other features in connection with this installation which are worthy of publication, the electrical end presenting a most unusual problem. The writer sincerely trusts the details will be placed in the technical records of the Am. Inst. E. E.

The only criticism to be made of the report is that, in the opinion of the writer, it should have been submitted, with results given in the metric as well as the standard system according to the precedent set by him on the test of the 30,000-kw turbine installed at 74th Street and referred to by the author. The writer has found by experience a far broader appreciation by foreign engineers of

work done and information thus given, and to his mind this is most desirable in our new and closer international relations.

O. H. Bathgate. The writer notes that the water rates reported in the paper correspond very closely with those obtained on the other group of 30,000-kw. turbines reported in tests run by Messrs. Stott and Finlay, five years ago, by allowing usual corrections for difference in steam pressure and temperature. From this conclusion it is therefore hardly reasonable to expect any further increases in turbine efficiency for this size unit unless obtained from units of radically different design.

As the writer is informed, the first group of units were designed for a maximum load 2000-kw. less than the second group, yet their maximum efficiency is obtainable at approximately the same kilowatt load under conditions as tests were run.

From an operator's standpoint, it would be interesting to know the water rate obtainable on this 30,000-kw. steam turbine under daily operating conditions at full load, and at load carried for maximum efficiency. A heat balance of the plant may be in use for the determination of water rates under acutal daily operating conditions.

Oscar F. Junggren. In Fig. 7 of the paper is a dotted line which shows that lower water rates were obtained when the turbine valve was blocked. The reason for this is that the secondary valve opened too early when the capacity of 22,000-kw. was reached, the machine being designed to have this valve open at 24,000-kw. capacity. Considerable improvement in water rates between 22,000- and 26,000-kw. loads can be obtained by setting the operating cam to open the secondary valve later.

H. B. Brydon. It is surprising to find pressures as low as 150 lb. in a plant operating with superheated steam.

In the layout of the plant there seems to be no valve between condenser and vacuum pump and yet the customary arrangements for testing the pump presumably on closed suction are provided.

Water-sealed glands are used which obtain their water supply from the condensate discharge. It would be interesting to know the original source of sealing-water supply. It would also be interesting to know the method of cleaning oily condensate.

There seems to be no provision for controlling the excess auxiliary steam delivered to the sixteenth stage of the main turbine.

With no provision for shutting off this additional steam and no governor control there will be danger of a run-away.

It seems to the writer that trouble will be experienced because no air washer has been provided for the ventilating equipment.

The writer would like to know the length of life of the rubber expansion joints of the condenser. It has been his practice for many years to use corrugated copper sheets with which no trouble has been experienced. The copper joint takes up more room than the rubber but it seems perfectly possible to make a copper joint take up little room and still have flexibility.

It seems to the writer that the hydraulic pump installed in in connection with the support of the condenser is not worth its cost. If the fatigue of the spring is to be measured by the pump in connection with the pressure gage it would seem that greater accuracy might be obtained by using micrometers.

The writer dislikes to see the vacua reported in inches of mercury referred to a standard barometer, preferring the use of absolute pressures.

OSCAR F. JUNGGREN answered some of Mr. Brydon's queries as follows: Condensate is used in the water seals in order to save power. There is also an emergency connection with the city water supply.

An automatic valve in the line supplying the excess auxiliary steam to the sixteenth stage of the main turbine is tripped and closed in case the turbine speed is excessive, thereby preventing run-away from this cause.

The use of rubber joints was developed by the Interborough Company and is standard practice; although in this installation they are not particularly necessary.

Because of the difficulty of predicting whether the condenser casing will be sufficiently flexible to make a spring support unnecessary it is the practice of the General Electric Company to support condensers on springs.

T. E. KEATING. The author is to be congratulated for his contribution to the Society, recording so completely the excellent performance obtained from his turbine and condenser installation.

The danger of admitting auxiliary exhaust steam to a low-pressure stage of the turbine has been referred to, as there is possibility of overspeeding, unless the flow is under governor control. The author's tests indicate that approximately 18,000 lb. of boiler

steam are required for operating the turbine at no load. The total steam consumption of two circulators, one air pump and one condensate pump is approximately 25,000 lb. This quantity of steam at atmospheric pressure, admitted to the sixteenth stage of the main turbine, would not it itself be sufficient to maintain the turbine at normal speed, but means should be provided for cutting off this flow before a dangerous overspeed is reached, due to other causes.

Vacuum curves are always of interest, due to the fact that there is but little reliable test data on this subject. The author does not state whether the correction curve used in Fig. 10 was obtained from an analysis of the turbine design or by means of a series of tests at different vacua. Theoretically, one would figure a break in the curves similar to that shown on the upper one, but probably test data would indicate a gradual change in characteristic corresponding to the water-rate curve in Fig. 7. An explanation of the manner of determining the speed curve corrections would also be of interest.

The method of determining the B.t.u. transferred per hour in the condenser should be explained. Calculations outlined in the following tabulation indicate heat transfer values of 294, 344, and 347 B.t.u. for Tests 2, 3 and 4, while Table 2, in the report, shows 322, 368, and 378 B.t.u. respectively. In arriving at the heat content in the exhaust steam, the writer has used the Rankine cycle efficiency ratios shown in Fig. 8. While the steam inlet and vacuum conditions on the condenser tests differed slightly from the turbine tests, there is not sufficient variation to affect the efficiency ratio appreciably For the heat losses in the generator, turbine and bearings, values of 9, 6, and 5 per cent for tests 2, 3, and 4 respectively, were used.

Test Number	2	3	4
Steam per hour, lb.	180,666	317,000	373,666
Heat content of steam at turbine throttle, B.t.u.	1,228	1,230	1,233
B.t.u. per lb. to condenser using efficiency ratios of Fig. 8 and allowing for generator and turbine losses	933	945	961
Temperature of condensate, deg. fahr.	87	91.3	91.5
B.t.u. per lb. steam absorbed by circulating water	878	886	902
B.t.u. transferred per hour	159,000,000	281,000,000	336,000,000
B.t.u. transfer per sq. ft. surface per hour per deg. mean temperature difference	294	344	347
Rise in temperature of circulating water	4.8	9.8	11.3
Circulating water, gallons per minute	66,000	57,300	59,400

GEORGE A. ORROK. Condensers are not usually designed for heat transfer conditions but for maximum conditions. Consequently for about 8000 hours a year they are probably running at less than half capacity. The maximum hot water temperature which used to be 74 or 75 deg. is now as high as 85 to 86 deg. due to the number of power stations delivering hot water into the East River.

Ever since large condensers have been built there has been trouble with the expansion joint or spring supports. The expansion joints are made tight with difficulty and spring supports are equally troublesome, but one or the other must be used unless the condenser is hung from the foundation supporting the turbine in which case the turbine-condenser joint is rigid and expansion may be forgotten. Very much smaller and less troublesome expansion joints in the circulating water and other pipes will, of course, be necessary in this case.

THE AUTHOR. Mr. Finlay's discussion is a most valuable addition and will clear up a great many of the questions which have been brought up in some of the other discussions. The use of the metric system along with the British system would no doubt have been desirable.

Mr. Keating calls attention to the fact that steam from the auxiliaries would not necessarily cause overspeeding of the turbine. This would be true if the auxiliary exhaust system of each unit was separated from the other units. In view of the fact that the auxiliary exhaust connection on the main turbine is connected to the entire auxiliary exhaust system of the station, and thus has almost an unlimited supply of low pressure steam available, it is desirable to equip this connection with an automatic valve. As pointed out by Mr. Junggren, a valve of this nature has been provided. In addition to this safeguard, the vacuum breaker is automatically opened upon overspeed.

The vacuum correction curves which are reproduced in the paper were furnished by the builders of the turbine and the author is advised by Mr. Junggren that they have been checked by actual tests. It will be noticed that the actual conditions were very near standard and so there was very litle correction necessary.

The discrepancy which Mr. Keating has pointed out in connection with the heat transfer in the condenser is due to the fact that the condenser tests were run before the final completion of the turbines. At the time of these tests, the turbine was being operated

without the 19th and 20th wheels, which of course had a very appreciable effect upon the heat content of the auxiliary steam. It is only fair to the condenser performance to call attention to the fact that the condenser was being loaded slightly above normal conditions due this excess heat in the exhaust steam.

Mr. Keating has used the values of 9, 6 and 5 per cent for the losses in the generator, turbine and bearings. The flat value of 6 per cent was used in computing the results as reported in Table 2.

As pointed out by Mr. Bathgate, there has been very little improvement in turbine design in so far as efficiency is concerned, since the installation of the three turbines at the 74th Street Power Station. However, without doubt many improvements have been made in mechanical details, etc. This emphasizes the great need for serious thought toward other means of converting heat energy into mechanical energy, as apparently the thermal efficiency of 25 per cent is all that can be expected with the steam turbine. At the present time, the mercury boiler and turbine seem to be the only means by which any marked improvement in economy can be accomplished.

Referring to the units at the 74th Street Power Station, it may be stated that these three turbines were designed for a maximum capacity of 30,000-kw. However, they were able to carry a maximum of 32,500-kw. at the time of the tests which were reported by Messrs. Stott and Finlay. The three machines described in this paper were designed for a maximum capacity of 35,000-kw. Judging from the pressure curves in Fig. 19, it is evident that loads in excess of this could be carried. However, the governors are set so that the maximum load will not exceed approximately 35,000-kw. Mr. Bathgate calls attention to the fact that the maximum efficiency is obtained at approximately the same load for both sets of turbines. This is not quite correct as the best load of the 74th Street machines is approximately 26,500-kw. while for the 59th Street machines it is 24,000-kw. This is of some advantage as the average load actually carried is nearer 24,000-kw. As to the water rates of these turbines obtainable under daily operating conditions at poor load and at the most efficient load, the author wishes to call attention to the fact that the turbine during the tests was operated under actual operating conditions and so the curve in Fig. 6 gives the water rate which should be obtained under actual operating conditions at any particular load.

Referring to Mr. Brydon's discussion there is a valve provided in the line between the condenser and dry vacuum pump. However,

this valve was unintentionally omitted from the diagram shown in Fig. 2.

Under normal operating conditions, the glands receive their water supply from the condensate discharge, which arrangement does not complicate the measurement of the condensate by Venturi meters which have been provided. As pointed out by Mr. Junggren, an emergency connection is also provided. However, this connection is with the condensate heater and not with the city supply.

It is supposed that Mr. Brydon refers to the oil in the condensate from the reciprocating engines, as of course the condensate from the turbine is absolutely oil free. A large portion of the oil from the reciprocating engines is removed in the receiver between the engines and the low-pressure turbines. A full description of this equipment will be found in the paper by Messrs. Stott and Pigott, reported in TRANSACTIONS vol. 38, p. 655.

As previously pointed out, the connection for the injection of auxiliary steam into the 16th stage, is provided with an automatic stop valve.

In regard to the use of air washers, the Interborough Rapid Transit Company feels that in veiw of its experience and tests, the use of the present type of air washer is not warranted in so far as the Interborough plants are concerned.

The company's experience with rubber expansion joints has been most satisfactory, as joints of this type have been in use at the 74th Street Power Station for the past seven years with practically no maintenance. However, some trouble was experienced with joints which were purchased at a later date. It is believed that this trouble was due to defective rubber which was furnished during the war period.

As to the flexibility of copper joints, it has been found that flexibility does not exist to the degree which is required in the case of large joints of this type. In fact, it was due to this lack of flexibility which drove the Interborough Rapid Transit Company to the development of the rubber type of joint.

Mr. Brydon does not seem to think that the hydraulic jacks as installed are desirable. The use of micrometers for detecting the fatigue would be rather uncertain due to the possible uneven settlement of the turbine and condenser foundations, while the jacks provide a means for determining the actual load irrespective of any variations such as foundation settlement.

No. 1792

AN INVESTIGATION OF OXY-ACETYLENE[1] WELDING AND CUTTING BLOWPIPES

By R. S. Johnston,[2] Washington, D. C.
Member of the Society

This paper reports the results of and conclusions from an elaborate series of tests carried out at the Bureau of Standards Washington, D. C. on commercial apparatus for cutting and welding by the oxy-acetylene process, submitted by manufacturers for the purpose of the tests.

The test equipment used consisted of a weighing system for determining the weight of the gases used, a system for metering the gases as a check upon the weight and for maintaining standard conditions of pressure and temperature, a welding table for the welding tests, a cutting table for the cutting tests, and a "flashback" and safety testing apparatus for determining the conditions of the flashback phenomena.

Unusual precautions were taken to insure accuracy in all the tests. For example, the cutting tests were all made by a mechanically controlled cutting device installed on the cutting table, to eliminate the personal equation.

The welding tests were performed upon ½-in. and ¾-in. steel plates, and the cutting tests upon ½-in., 2-in., 6-in. and 10-in. material.

The general conclusions from the tests were that there was a great deal of difference between the characteristics of different designs of cutting blowpipes, and that there was no make of apparatus which was equally proficient and economical for all thicknesses of metal. Further, one of the prime essentials of a good welding blowpipe is its so-called gas ratio, which should be unity. Not any of the blowpipes tested proved capable of maintaining a gas ratio of unity during welding although, as the author states, the welds were probably made with greater care than has ever been bestowed upon any like work.

The important problem of "flashback" receives extensive consideration, and the author concludes by stating that experience gained during these tests indicates that a blowpipe designed to be absolutely free from flashback caused by any form of obstruction, under all working conditions, will also be the eminently safe blowpipe and the one which with ordinary care will produce sound welds. Such a blowpipe will be one so designed that, under all conditions of operation—even to complete blocking of the gas exit at the tip end, there will be maintained a one-to-one volume delivery of each gas, at identical pressures.

[1] Published by permission of Dr. S. W. Stratton, Director Bureau of Standards, Washington, D. C.

[2] Engineer-Physicist, U. S. Bureau of Standards, Washington, D. C.

Presented at the Spring Meeting, Chicago, Ill., May 23 to 26, 1921, of The American Society of Mechanical Engineers, New York.

IN the war the increased use by the American Expeditionary Forces of oxy-acetylene welding and cutting equipment necessitated large purchases by the Government. In discussions concerning the relative merits of procurable apparatus it became evident that no authentic data were available as to the relative merits of the various blowpipes. The Bureau of Standards was therefore requested by the Chief of Ordnance to make a test for the determination of "efficiency, safety and workmanship entering into the several makes of apparatus" (oxy-acetylene).

2 Of necessity, the emergency of the investigation limited its proposed scope. The manufacturers of welding and cutting blowpipes were invited to conference and were circularized in regard to methods of test. From a study of the results of these conferences and correspondence a tentative scheme of tests was developed. A series of preliminary tests was started, and from the results a set of tests and a method of conducting them were decided upon. Further study made it desirable that a more extended investigation of the oxy-acetylene blowpipe be made. The signing of the armistice relieved the urgency and a much more complete series of tests was proposed. The results of these tests are included in this paper.

3 In deciding on the final tests S. W. Miller, Mem. Am. Soc. M. E., was engaged as consulting engineer. After securing his suggestions the tests were submitted to the War Department for final suggestions before being submitted to the manufacturers of the apparatus to be tested.

4 Several weeks of preliminary work were given over to acquainting the expert welders and cutters from the Naval Gun Factory, Washington, D. C., and the New York Navy Yard with their new duties. The knowledge that the tests were to be started, together with the fact that most of the apparatus had been held for test for at least a year, brought forth requests from the manufacturers for the privilege of submitting new and improved apparatus, which were granted.

5 A copy of the tests to which it was proposed to submit blowpipes, accompanied by a circular letter giving the sizes and chemical analyses of the materials to be cut and welded, was forwarded to each manufacturer for him to furnish certain information. Each concern was later also individually notified of the day upon which its apparatus would be tested.

6 The tests occupied about three months.

ACKNOWLEDGMENT

7 In the development of this investigation the general experience of a number of people was drawn upon that the question might be viewed from its broadest aspects, and its successful completion was largely due to the splendid coöperation received. Appreciation for such assistance is expressed to Col. L. B. Moody, Ordnance, War Department; Lt.-Col. Warren R. Roberts, Q. M. C., War Department; Maj. A. B. Quinton, Jr., Ordnance, War Department; Maj. R. F. Carr, U. S. A. War Department; Maj. W. L. Simpson, Ordnance, War Department; Capt. A. L. Willard, Naval Gun Factory, Washington, D. C.; Capt. G. H. Rock, Hull Division, New York Navy Yard; and to the officers of the divisions of the Bureau of Standards and the several manufacturers of oxy-acetylene equipment.

8 The paper presented herewith is the result of a special investigation conducted at the Bureau of Standards for the War Department, as represented by Major A. B. Quinton, Jr., Tank, Tractor and Trailer Division, Ordnance Department, through which Department the results of the investigation have been made available for public distribution.

SUMMARY OF RESULTS

9 The results of this investigation would seem to warrant the following statements:

For the Cutting Blowpipes:

a That there is today no generally accepted theory for proportioning, for the cutting of metal of various thicknesses, the volume and velocity of the issuing cutting jet, with the result that none of the apparatus submitted to test proved economical for all thicknesses.

b That there is for any thickness of metal cut a limiting velocity of exit of the cutting jet at which complete utilization of the oxygen takes place and a limiting value for the amount of oxygen required to produce a cut.

c That an increase in acetylene consumption, of oxygen consumption, or of the velocity of exit of the cutting jet beyond the limiting values, does not produce increased efficiency in commensurate ratio.

d That a large majority of the blowpipes tested were equipped with excessive preheating flames for the thickness of metal the tip

is specified for, and that such excessive-sized flames are disadvantageous both from the standpoint of economy of operation and quality of work performed.

e That considerable improvement in economy of operation seems possible in cutting material of 2-in. thickness and that possibly this condition may be found to exist for metal of other thicknesses than those used in the tests.

f That the maximum thickness of metal that may be economically cut with an oxy-acetylene blowpipe of standard design when neither the material nor the oxygen is preheated and the cutting is done only from one direction, is about twelve inches.

g That cutting blowpipes due to their incorrect design are subject to the same "flashback" troubles found in the welding blowpipes.

For the Welding Blowpipes:

a That the blowpipes most subject to the so-called phenomena of flashback are those in which the oxygen is delivered at a pressure in excess of that at which the acetylene is delivered.

b That all the blowpipes tested, including those in which the acetylene is delivered at an excess pressure as well as the so-called equal- or balanced-pressure blowpipes, are subject to flashback phenomena on account of inherent defects in their design.

c That the cause of the development of the conditions producing flashback is the setting up within the blowpipe tip and head of a back pressure which retards or chokes off the flow of one of the gases.

d That this back pressure is the result of confining or restricting the volume flow of the issuing gases at the tip end.

e That any cause tending to restrict the flow of the gases sets up a back pressure which immediately causes a change in the amount of each gas delivered to the mixing chamber.

f That a fluctuating gas-volume ratio, due to restriction of volume flow, from whatever cause, prevents a blowpipe from maintaining constantly and at all times during operation the desired "neutral flame."

g That a blowpipe that cannot maintain under all operating conditions a neutral flame cannot logically be expected to produce sound welds.

h That all the blowpipes tested during this investigation either through improper gas pressures or improper interior design or both are incapable of maintaining a neutral flame (constant-volume gas

ratio) under all conditions of restricted gas flow and are therefore incapable of producing sound welds where there is any liability of the gaseous products of combustion being momentarily confined, such as occurs in practically all welding operations.

i That the ability of a blowpipe to consume an equal volume ratio of gases when burning freely and undisturbed in air is no criterion

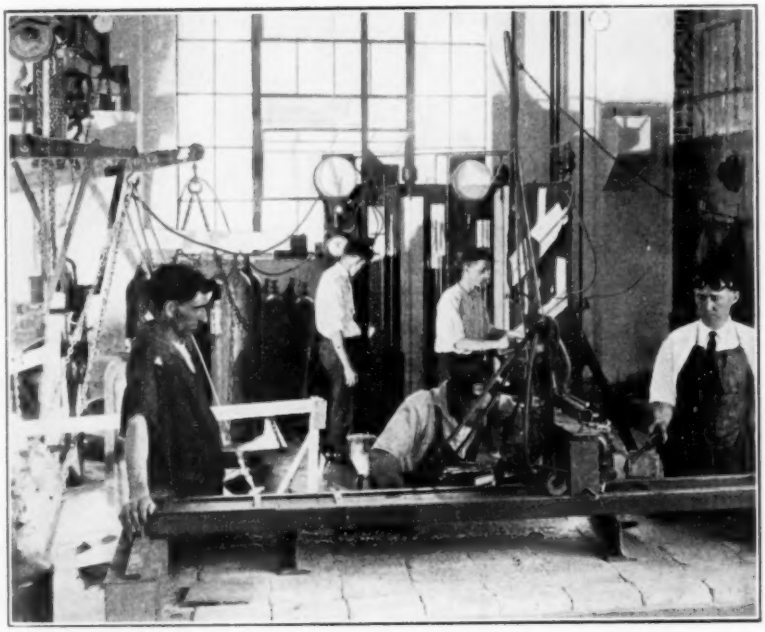


FIG. 1 ENTIRE TESTING EQUIPMENT

that it is capable of producing sound welds, i.e., that it is not subject to detrimental fluctuations in gas ratio during a welding operation and therefore is capable of maintaining a neutral flame under all operating conditions.

j That whether a blowpipe of present designs will consume an equal volume ratio of gases when burning freely and undisturbed in air depends on how nearly correct the operator sets the so-called "neutral" flame, and experience indicates that the average operator checks the acetylene gas flow too much and actually develops an oxidizing rather than a neutral flame.

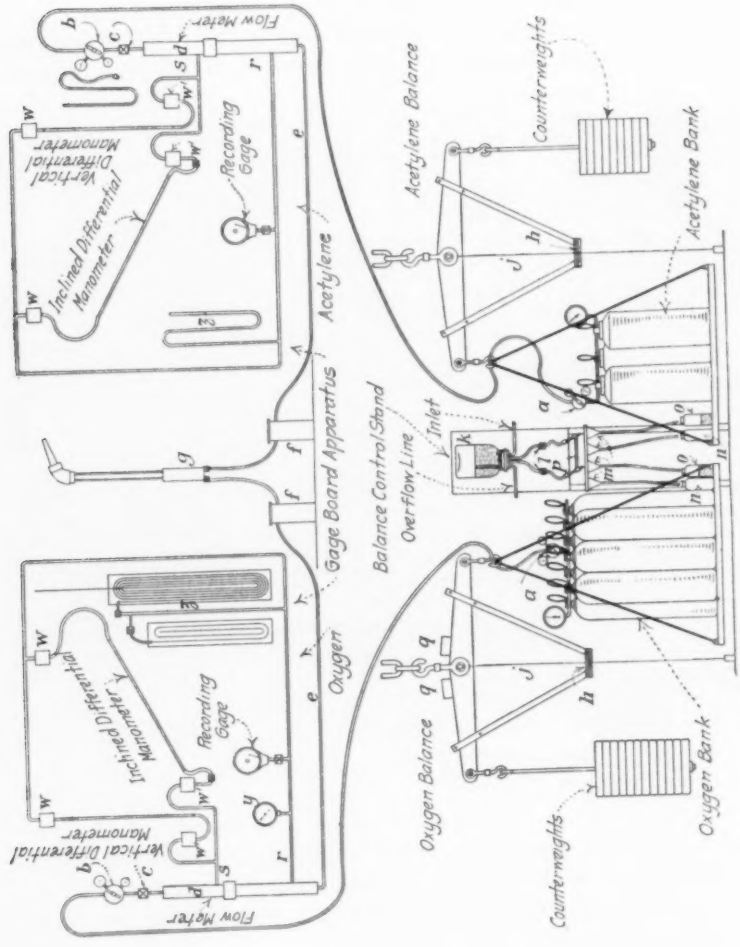


FIG. 2 DIAGRAM OF ENTIRE TESTING EQUIPMENT

k That the question of the possible limiting strength and ductility or the efficiency of welds made by the oxy-acetylene welding blowpipe must await the development of a more satisfactory instrument, and that having such an instrument there is no reason to believe that a weld of clean, sound metal cannot be made with

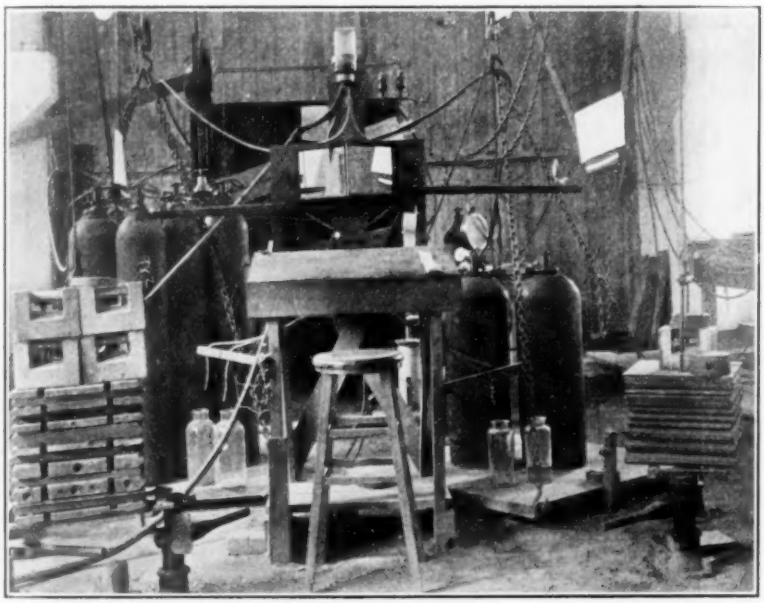


FIG. 3 OPERATOR'S TABLE AND BALANCES

assurance during any welding operation and that such welds will or can be made to possess the proper physical properties.

DESCRIPTION OF EQUIPMENT USED

10 In general the equipment used for making the tests may be listed as:

a Weighing system for determining amount of gases used during tests by loss of tank weight

b Gage-board system containing necessary pressure gages, regulators and orifice flowmeters

c Welding table
d Cutting table
e Flashback and safety testing apparatus.

11 Figs. 1 and 2 show the entire equipment. The tanked gases are "banked" and counterpoised on an equal-arm balance.

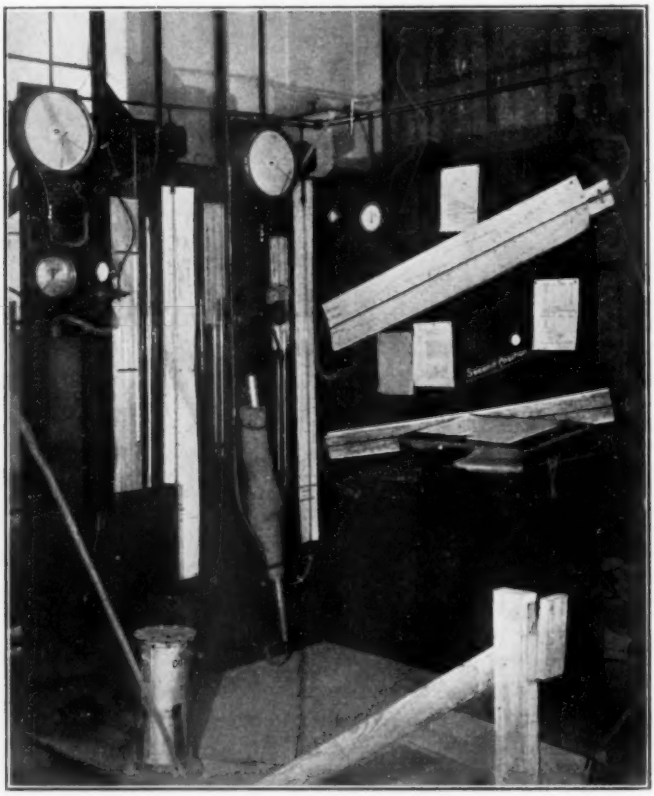


FIG. 4 GAGE-BOARD SYSTEM

The gas from these tanks passes through a regulator *a*, thence through a flexible hose to the back of the gage board. Passing through the board the supply line enters a second regulator *b*, thence through a needle valve *c*, to the top of and through an orifice flowmeter *d*. The gas coming from the extreme bottom of the flow-

meter is then conducted through a standardized length of flexible hose *e*, containing a safety flashback tank *f*, to the blowpipe to be tested.

12 Fig. 3 is a view of the operator's table, showing the use of

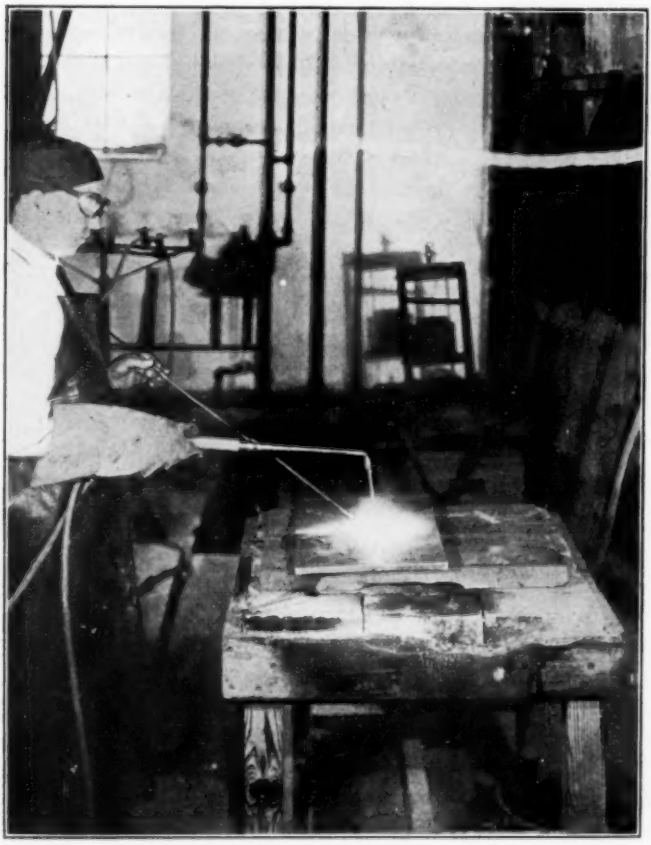


FIG. 5 WELDING TABLE

mirrors for taking practically simultaneous readings of the balances. Fig. 4 shows the gage-board system. In the center of the board is one of the flowmeters, surrounded by wool-felt insulation.

ACCURACY OF TESTS

13 Unusual precautions were taken to insure accuracy.

14 If the gas losses through leakage exceeded 0.01 to 0.02 lb.

per hour, actual testing of the blowpipes was not continued until the leaks causing such losses were located and stopped.

15 The values obtained for gas consumption were accurate to 0.005 lb. and in most cases probably much closer.

16 With the precautions taken with the regulators, it was readily possible to maintain under almost all conditions a pressure varying not more than 0.01 or 0.02 lb. from the desired amount. Autographic records gave visual evidence of the absolute uniformity of the pressure under which the blowpipes were operated during the tests.

17 Special orifice flowmeters were designed to facilitate securing data on continuous blowpipe action and as a check on the weighing system.

18 To complete the records, the gage board was equipped with a standard calibrated thermometer, a psychrometer, a barometer and a stop watch.

19 Elaborate precautions were taken to maintain the gas conditions constant during the tests.

THE WELDING TABLE

20 All welding during the tests was performed upon the welding table illustrated in Fig. 5. This was a wooden-frame table approximately three feet square, the top of which was composed of firebricks. On top of the firebricks was placed a heavy casting channelled for a width of about six inches throughout its length. This formed the base upon which all the plates for welding rested during the welding operation. The plates were aligned centrally along this base with the idea that the casting with its grooved surface would permit of better heat radiation along the line of the weld and at the same time form a baseplate or background to prevent possible inconveniences from the blowholes caused by blowing the welded material through the bottom of the V of the test weld plates.

21 As indicated in Fig. 5, the line of the weld was placed directly in front of the welderand the welding was performed from the back toward the operator, thus giving him a full view of the work as it progressed. The welded plates were cut so that the welds were 1 ft. in length. Where 2 ft. of weld were made continuously, pairs of plates were set in front of each other with a slight space between the individual pairs and with proper allowance for expansion so

that the process could be carried from one plate to the other without any interruption. The groove along the baseplate facilitated the preheating of the second pair of plates, so that the start upon the second weld was made under practically the identical conditions which existed when the first pair of plates was finished, a condition



FIG. 6 MECHANICALLY OPERATED CUTTING TABLE

that would be equivalent to that which would occur if the weld was made as one of 2-ft. length instead of two of 1 ft.

THE CUTTING TABLE

22 In order to minimize as far as possible the personal equation entering into cutting tests, all tests were made by a mechanically controlled cutting device installed upon the cutting table, Fig. 6 and Fig. 1. This was a wooden-frame table approximately 4 ft. by 12 ft. in size with metal-bound edges and a firebrick top.

At each end of the table and securely fastened to it were 6-in. by 6-in. sticks placed crosswise, along the top of which were fastened 1-in. by 2-in. bars of metal. These latter acted as runners to carry a channel placed with flanges down as illustrated in the figures. The channel held the track upon which the mechanically controlled cutting device operated. Longitudinal motion of the blowpipe could therefore be secured by causing the machine to move along the track and lateral shifts were made by sliding the channel along the runner bars screwed to the 6-in. by 6-in. end blocks. In order to facilitate quick lateral movement of the channel and insure, where such was

TABLE 1 TIME SPEED RECORD FOR CUTTING ¼-IN. METAL—TORCH NO. 14

Length of cut, ft.	Length of time to make cut, min.	Time to cut each 5-ft. increment, min.
0	0.0	0.0
5	2.73	2.73
10	5.42	2.69
15	8.28	2.86[1]
20	11.01	2.73
25	13.78	2.77[1]
30	16.52	2.74
35	19.28	2.76[1]
40	22.03	2.75
45	24.88	2.85[1]
50	27.58	2.70

[1] It was found during the tests that 2 to 4 sec. were consumed in reversing direction of cut at end of each 10 ft. of length.

desired, that pieces of definite width could be cut, the runner bars had a series of holes spaced conveniently at 2 in. on centers and a set of metal pins which fitted these holes. See Fig. 1. By inserting the pins in the proper holes the channel could be instantly shoved over a definite required distance.

23 This compound arrangement facilitated greatly the making of continuous cuts of 10 ft. or upward. The cutting machine being fitted with a reverse-motion gear could be operated in a forward direction for the full length of the table, reversed in direction of travel and at the same time the channel slid over a definite distance and thus the blowpipe made to travel backward for the full length of the table for a cut on a new and adjacent line.

24 In order to furnish room for the disposition of the slag formed during the cutting operation, the metal to be cut was supported on a system of teeth as shown in Fig. 6. These teeth proved extremely satisfactory and made a very convenient method of maintaining metal in proper position for cutting.

25 A typical speed record is given in Table 1.

THE FLASHBACK AND SAFETY APPARATUS

26 *Flashback-Protection Tanks.* The testing equipment also included in the gas lines two flashback tanks (*f*, Fig. 2, and Fig. 1).

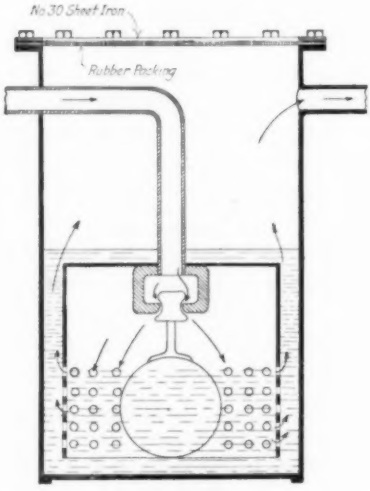


FIG. 7 FLASHBACK–PROTECTION TANK

These tanks were essentially hydraulically controlled valves which were intended to prevent the propagation of an explosion in the blowpipe or gas line backward toward the gas supply. They are shown in sectional view on Fig. 7. While it was generally realized that the installation of the water seal of these flash-protection tanks might be considered detrimental, due to the absorption of moisture by the gas, it became evident that their installation was nevertheless a prime necessity as a means of protecting the rather expensive gage-board equipment. It was believed, further, that, inasmuch as the oxygen in use generally came from cylinders that contained

more or less water, the passing of the gas through the hydraulic seal of the flash-protection tanks would in reality tend to standardize the moisture content in the gas and therefore produce similar effects for all blowpipes.

27 These flash-protection tanks proved extremely satisfactory for the purpose intended, in that in several explosions they prevented the propagation of the flame beyond the flash tank. They generally ruptured by the blowing off of the head of the tank during the explosion. As furnished the heads were of rather thick sheet metal, fastened on with bolts as indicated on Fig. 7. This construction proved to be somewhat dangerous to the operators making the tests and the tanks were therefore modified in their construction, as indicated in the figure, by having a rubber packing and a thin sheet of metal fastened to the top with a heavy annulus. By this construction it was expected that if an explosion developed within the flash tank, the thin metal sheet would rupture by tearing and thus minimize danger from flying parts.

28 *Flashback Apparatus.* The tests for freedom from flashback and safety in operation were conducted with the so-called flashback apparatus. For this series of tests a pair of flash tanks were connected into the hose lines immediately back of the blowpipe. Between the flash tanks and the blowpipe proper two observation boxes were installed. These observation boxes consisted of a glass observation tube inserted in the gas lines and protected by a glass-covered observation box. The box was of wooden side and back construction with a vented back and a duplex glass face. The interior of the observation box was painted dead black to facilitate observation of the flame propagation. This equipment was used throughout the entire series of flashback tests and for the severe flashback test mentioned later.

MATERIALS USED IN TESTS

29 *Welding Rod.* The welding rod used throughout the entire series of tests was secured from the Naval Gun Factory, Navy Yard, Washington, D. C. This rod was purchased in July 1917 under Navy Department Specification 22–W–4. A number of chemical analyses were made and the percentage composition was found to be as follows:

Carbon	0.024 to 0.03	Silicon	0.002 to 0.004
Manganese	0.05 to 0.08	Chromium	Trace
Phosphorus	0.01 to 0.015	Nickel	Not detected qualitatively
Sulphur	0.023 to 0.024	Vanadium	

30 *Steel Plates for Welding and Cutting.* The steel plates used for welding were $\frac{1}{2}$ in. and $\frac{3}{4}$ in in thickness. The material used for cutting was $\frac{1}{2}$, 2, 6 and 10 in. in thickness. All the material used in both welding and cutting, except the 10-in., was furnished through the Engineer Corps, War Department, and was selected with special reference to uniform quality for any particular thickness. The $\frac{1}{2}$-in. material was furnished in plates 3 ft. by 5 ft. in size and was used for both welding and cutting tests. The middle section of each plate was retained as a sample for determining the qualities of the plate. The remaining pieces were used for making welds. During the welding tests it was the practice to use plates that were adjacent to each other in the main or full plate before it was cut into weld specimens, so that as nearly as possible the material used for any particular test would be identical.

31 The $\frac{3}{4}$-in. material for welds was received in plates 12 in. wide by 6 ft. in length. These plates were cut up into sections 9 in. in length, and for the full width of the plate, that is, 12 in. All specimens for welding tests were finished with a butt joint of the single V 90-deg. included-angle type.

32 For the cutting tests the 3-ft. by 5-ft. by $\frac{1}{2}$-in. plates were cut into strips approximately $1\frac{1}{2}$ to 2 in. in width as indicated in the description of the cutting apparatus above. The 2-in. material for cutting was furnished in sections 2 in. by 6 in. by 20 ft. These were cut, for convenience in handling, into 5-ft. lengths and in test operations cut lengthwise into sections of 2 in. width. The 6-in. material was shell billet steel furnished in 3-ft. lengths and was cut lengthwise in test operations.

33 Chemical analyses of these materials indicated that they were of approximately the following percentage compositions:

$\frac{1}{2}$-in. mild-steel plate for welding and cutting tests:		$\frac{3}{4}$-in. plate for welding tests:	
Carbon	0.14	Carbon	0.25 to 0.27
Manganese	0.32 to 0.36	Manganese	0.41 to 0.48
Phosphorus	0.012 to 0.013	Phosphorus	0.011 to 0.013
Sulphur	0.033 to 0.055	Sulphur	0.041
Silicon	0.006 to 0.012	Silicon	0.004
2-in. mild steel for cutting tests:		6-in. steel for cutting tests:	
Carbon	0.19 to 0.20	Carbon	0.46 to 0.51
Manganese	0.42 to 0.44	Manganese	0.60
Phosphorus	0.025 to 0.027	Phosphorus	0.029 to 0.038

Sulphur	0.055	Sulphur	0.064 to 0.067
Silicon	0.009 to 0.017	Silicon	0.13 to 0.14

34 For extra heavy cutting it was desired to determine what the various blowpipes could do on special material approximating armor plate in quality. For this purpose the Naval Gun Factory supplied a cast-steel billet 10 in. by 36 in. by 50 in. in dimensions. This proved to be of the following percentage chemical analysis:

Carbon	0.38	Silicon	0.25 to 0.28
Manganese	1.21 to 1.28	Copper	0.96
Phosphorus	0.038 to 0.043	Nickel	2.62 to 2.70
Sulphur	0.019 to 0.027	Chromium	Less than 0.01

DESCRIPTION OF THE TESTS

35 The tests were started with the idea of submitting each manufacturer's equipment to the series of tests listed in a circular sent out under date of February 18, 1920. It was found, however, that the proposed series of tests was excessive from the time standpoint. One of the most serious drawbacks as a time-consuming element was the fact that a very large percentage of the blowpipes submitted for test would not operate with the pressures specified by the manufacturers. This condition was probably due to the fact that it is quite a customary practice to recommend setting the regulator pressures three to five pounds higher than the specified blowpipe pressures. By throttling the gases at the blowpipe-handle valves the operator insures having sufficient pressure available at all times to maintain the required velocity of exit of the gases at the tip end. He is therefore enabled to compensate for pressure fluctuations due to irregular action of the regulator, thus tending to minimize the development of flashbacks. The specifications for the tests distinctly stated that at least one of the blowpipe-handle valves must be maintained at full opening during a test. It was only by such a procedure that the gas consumption of a blowpipe could be definitely ascertained. For a great many of the blowpipes the pressures were too high to enable the maintenance of a stable flame with one of the handle valves at full opening.

36 Another quite serious source of trouble from the standpoint of time consumption was that due to leakage, necessitating the dismantling and repacking of valves.

37 In order, therefore, that the entire investigation might not

require an undue length of time it was decided that attention should be devoted only to the so-called primary tests, consisting of the welding, cutting, gas-ratio, and flashback tests. Such proposed tests as the variation of pressure within the blowpipe head, etc., were therefore abandoned. On the basis of the foregoing the following schedule of tests was adopted and all blowpipes tested during this investigation were submitted to them.

38 *Welding Tests.* All blowpipes reported upon were submitted to five welding tests, designated respectively as Tests 1*a*, 1*b*, 1*c*, 1*d*, and 2. All the tests numbered 1 were made with $\frac{1}{2}$-in. plate. Test 2 was a weld with $\frac{3}{4}$-in. plate.

39 Tests 1*a* and 1*b* were made with the tip sizes and pressures specified by the manufacturer if this were possible. For both of these tests a 2-ft. length of weld was made. These tests were identical in all respects, with the exception that an attempt was made to evaluate the personal equation by using different operators. For Tests 1*c* and 1*d* a 12-in. length of weld was made. Both of these welds were made by the operator who made the weld of Test 1*a*, the idea being to maintain as nearly constant a personal equation for this series of tests as possible. Test 1*c* was run with the same size of tip as 1*a*, but with pressures (both oxygen and acetylene) 50 per cent in excess of the pressures used for Test 1*a*. Test 1*d* was carried out similarly to 1*c*, except that the pressures were 25 per cent below those used in Test 1*a*.

40 As mentioned above, the pressures specified by the manufacturer very often gave an exit velocity to the gas too high to permit of maintaining a stable flame at the blowpipe tip. In such cases the manufacturer's representative was requested to furnish a modified pressure that would enable the maintenance of a stable welding flame. Very often the modified pressure thus determined upon would not permit of the application of Test 1*c*, that is, a test with a 50 per cent increase in pressure in both gas lines. It was customary, therefore, in such cases to modify the test procedure and incorporate as a test in place of Test 1*c*, Test 1*e*, which was run under identical conditions with the above test, with the exception that the pressure on both gas lines was reduced to 50 per cent of the pressure used to make Test 1*a*.

41 Tests 1*c*, 1*d*, and 1*e* were incorporated to show the effects of increased or decreased pressures on the operation and economy of the blowpipe. Such excess or decreased pressures are found to be quite common in many welding operations, due to carelessness

on the part of the operator in setting regulator pressures or to imperfect regulator action. It was felt that a properly designed blowpipe should be capable of adjustment over a considerable range for any specified tip size. It was hoped in the investigation to secure data that would either verify this assumption or prove that it was absolutely essential to maintain exact pressures for satisfactory blowpipe operation.

42 Test 2 was a 12-in. length of weld of $\frac{3}{4}$-in. mild-steel plate. This weld was made in all cases by the operator who made the weld of Test 1*b*. This test was selected as indicating the probable results to be obtained with a blowpipe in heavy welding, and with Test 1 it was felt that it would give a fair idea of the adaptability of the blowpipe for welding purposes. The tips for welding $\frac{1}{2}$- and $\frac{3}{4}$-in. plate were selected as being the tips used respectively for the average-size weld and for the maximum-size weld, and therefore the best general average for determining the blowpipe's efficiency and safety.

43 *Cutting Tests.* The cutting blowpipes reported upon have been submitted to eight cutting tests, four of which may be considered primary. In each of the cutting tests a length of cut was made that was sufficient to determine accurately the gas consumption of the blowpipe. Test 6*a* consisted of cutting continuously 50 linear feet of $\frac{1}{2}$-in. mild steel, Test 7 was a continuous cut of 10 ft. of 3-in. mild steel, Test 8 a continuous cut of 6 ft. of 6-in. medium steel, and Test 9 a cut by hand for maximum thickness.

44 Each test cut was preceded by a preliminary trial cut to determine the maximum speed at which the blowpipe would cut with the tip size and pressures specified by the manufacturer for the particular thickness of metal. It was found during these trial cuts that the same condition existed relatively to the cutting blowpipes that was found to exist with the welding blowpipes, that is, that a great many of the pressures specified by the manufacturers for cutting particular thicknesses of metal were not suitable for the work intended. One of the main troubles seemed to be that the majority of the blowpipes were operated with preheating flames of too high intensity.

45 Throughout the entire series of cutting tests all blowpipes were operated similarly, that is, at least one of the gas-control valves for the preheating flames was at full opening, the preheating flames were adjusted to neutral with the oxygen cutting valve at full opening, and all cutting operations were started with this latter

open. Some criticism of this procedure developed from time to time, but throughout the entire series of tests it was found that at no time were any detrimental results produced, and that even on cuts as heavy as 16 in. no trouble was found in starting with the oxygen cutting valve open. This method was adopted to insure the maximum capacity of the blowpipe throughout the entire test period, and further to enable the maintenance of accurate pressures throughout the test and insure accurate data as to the maximum gas consumption during the cutting operation.

46 Test 9, cutting for maximum thickness, was originally conducted with the idea of submitting the blowpipe to a test for the maximum thickness specified by the manufacturer as permissible for his blowpipe. It was found during the course of the investigation that the great majority of the blowpipes would not successfully cut the so-called maximum thickness specified by the manufacturers and published in their literature. Experience during the investigation seemed to indicate that the probable maximum commercial limit for economic heavy-cutting operations when the cut is made in one direction only with an ordinary stock blowpipe, lay in the neighborhood of 10 in. It was therefore decided that all blowpipes should be submitted to the same maximum or heavy cut. A cut in material 10 in. in thickness and 12 to 18 in. in length was therefore made as a supplementary heavy-cut test for each blowpipe.

47 The tests for maximum thickness of cut were made by hand. On account of the possibility of damage to the gage-board equipment caused by heavy slag showers from pockets, etc., produced during cutting, this test was conducted out of doors. The original tests for maximum thickness were therefore made on different days, with the temperature of the metal varying from 35 to 68 deg. After the decision had been made to submit all blowpipes to cutting tests of identical thickness, it was decided to minimize the effects of varying temperature as much as possible on the heavy cut by making all cuts during the course of one or two days. By this procedure only a slight temperature change occurred. Particular care was also exercised to insure that a blowpipe was not submitted to test on pieces of the steel ingot used for this work that were still warm from a test of another blowpipe. The test for heavy cutting, that is, the special 10-in. cut just described, is listed in the data that follow as Test 9.

48 Besides the primary tests it was thought desirable also to determine the efficiency of the blowpipe as it was affected by varia-

tions in the oxygen pressure supplied. For this purpose a series of supplementary tests listed as Tests 6*b*, 6*c*, 6*d*, and 6*e* were conducted. These were made on $\frac{1}{2}$-in. mild-steel plate and a cut of 10 linear feet made for each test. In this entire series of tests the tip sizes and the conditions of cutting were identical with Test 6*a* listed above, with the exception that the oxygen pressures alone were modified. These were increased 50 and 25 per cent and decreased 25 and 50 per cent, respectively, for Tests 6*b*, 6*c*, 6*d* and 6*e*. No change was made in the acetylene pressure. A slight aspirating effect for the increased oxygen pressure was expected and was shown in the volume of the acetylene used.

49 *Gas-Ratio Tests.* One of the prime essentials of a good welding blowpipe is its so-called gas ratio, that is, the ratio of the volume of oxygen to the volume of acetylene consumed. Theoretically a properly adjusted blowpipe requires equal volumes of both gases, giving the ratio of 1 to 1. In order to establish the ratios of the blowpipes under test, each one reported upon was submitted to a series of gas-ratio tests, numbered 5*a*1 and 5*b*.

50 For these tests the blowpipe was allowed to burn freely in air with the same tip sizes and pressures as were used in the welding tests mentioned above. The blowpipe was supported upon a bracket stand with the tip in horizontal position. All gas-ratio tests were made upon the tips used for welding and no attempt was made to clean them before making the tests other than to blow them out with a rather high oxygen pressure. These tests might be expected to indicate the gas consumption of the blowpipe with the flame burning undisturbed from accidental obstructions, such as slag adhesions to the ends of the tips, etc., which occur in actual welding practice. The discussion of the welding blowpipe furnishes additional interesting information concerning the gas-ratio tests.

51 *Flashback Tests.* In order to determine the probable safety of operation of a welding blowpipe and, further, to secure information concerning the permanency of construction of the tip and blowpipe head, the blowpipes reported upon were submitted to two types of flashback tests.

52 One series of tests designated 3*a* and 3*b*, flashback tests on the tips used for welding $\frac{1}{2}$-in. metal and $\frac{3}{4}$-in. metal, respectively, consisted of the standard series of tests used by the Underwriters' Laboratories for determining the freedom from flashback and the safety of the welding blowpipe. Each of these tests consisted of four distinct operations. The first three of these operations were

carried out as follows: After being properly adjusted to neutral flame the blowpipe was tested for flashback by drawing the tip at varying angles across the surface of, and finally pressing the tip end firmly against, certain materials. For this test a cold steel plate, a firebrick, and a piece of wood were used. Finally the tip was used to make a pool of molten metal in a cast-iron block, flux being used to assist in maintaining the fluid condition of the metal and the tip suddenly plunged into the pool of metal.

53 Another series of tests intended to determine the permanency of construction and designated as the Severe Flashback Test, Test 4, was carried out by supporting the blowpipe on a vertical sliding carriage in such a manner that at the proper instant the ignited and carefully adjusted blowpipe could be lowered so that the end of the tip was directly over the center of a hole 2 in. in diameter and 2 in. deep, drilled in a heavy cast-iron block.

54 In all the flashback tests the blowpipe was operated under the identical conditions at which it was supposed to be used during the welding operations, and after the initial adjustment of the flame no further adjustments were permitted. Following also the procedure in the welding tests, at least one of the blowpipe-handle valves controlling the gas had to be at full opening.

55 *Log-Sheet Records of Tests.* It has not been thought advisable, in the light of the discussions following, to incorporate as parts of the report the test log sheets, especially as there are over 1900 of them.

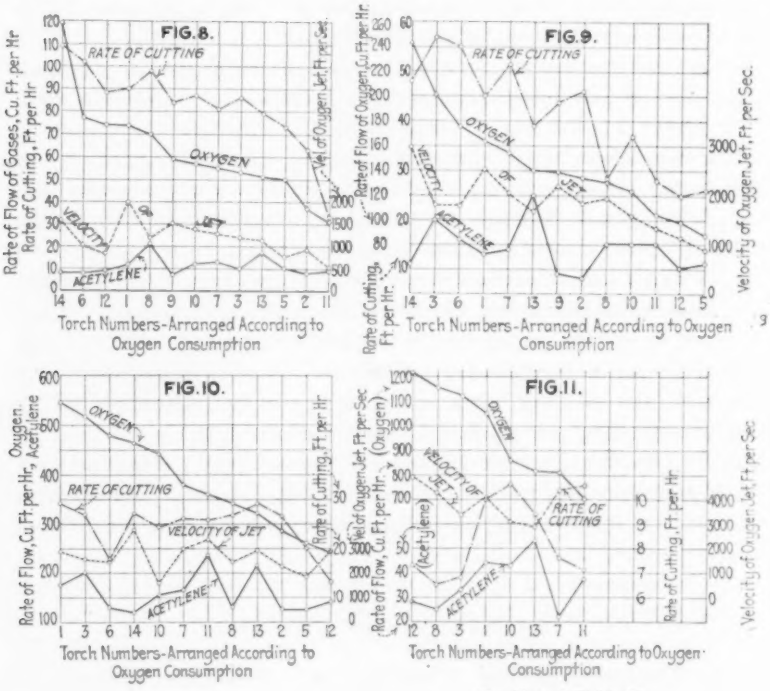
DISCUSSION OF TEST RESULTS

56 *General Basis of Consideration.* Summaries giving the results obtained with the various blowpipes and upon which the following discussion of test results is based are included in curves of Figs. 8 to 15 and in Tables 2 to 6. For detailed study, reference may be had to the tables.

57 For convenience in analysis the results of these tests are summarized in a series of plotted graphs. In the plotted summaries are shown the results of tests on blowpipes from fourteen different manufacturers, each represented by a designating number. The group submitted to test comprises most of the better-known and more widely advertised makes of apparatus on the American market and includes besides those voluntarily submitted three makes of apparatus that were withdrawn by their manufacturers because of

disagreement over methods of test procedure. Samples of these latter three were purchased for test purposes.

58 For certain studies on the probable economies of operation of the various blowpipes it was necessary to assume certain fixed values as units of compensation and cost. While this procedure may be open to certain criticisms due to the fact that the assigned



(See Note 4, Table 2)

FIGS. 8 TO 11 RELATION OF RATE OF FLOW AND VELOCITY OF EXIT OF OXYGEN TO RATE OF CUTTING METAL

(Fig. 8, for ½-in. metal; Fig. 9, for 2-in. metal; Fig. 10, for 6-in. metal; Fig. 11, for 10-in. metal)

values are arbitrary and may not apply under conditions of operation ordinarily met with in commercial work, still it is only through such economy studies that certain features of blowpipe operation can be distinctly shown. For such studies the standard salary schedule of the U. S. Naval Gun Factory, Washington, D. C., for the summer of 1920 was selected as the basis of compensation for labor costs. On this assumption the labor costs for welding and cutting have been computed at 80 cents per man-hour. For specify-

ing the value of the gas consumed during any particular operation the prices of $1.50 per 100 cu. ft. for tanked oxygen, and $2.75 per 100 cu. ft. for tanked acetylene, were assumed.

59 Before definitely settling on the above prices, cost data for several of the test series were computed for varying ratios of cost of oxygen to acetylene. These latter figures showed that no change in rating occurred in the relatively more and relatively less economical blowpipes, and that such changes in rating as did occur took place in the few that seemed to be rated between these two groups, changes which in reality were of no importance.

60 In the computations of cost of operation only gas and labor costs are included. No additions have been made for maintenance of equipment, overhead charges and secondary supplies.

61 Certain of the plotted data contain curves showing the relative velocity of exit of gases from the blowpipe tips. In order to plot these curves for the cutting blowpipes it was assumed that the volume of oxygen consumed by the preheating flames was equal to the measured volume of acetylene consumed and that the volume of oxygen available for the oxygen cutting jet was therefore the total measured volume of oxygen consumed, minus an amount equal to the measured volume of acetylene consumed. For the welding blowpipes the velocity of exit was based upon total volume of both gases consumed.

62 *The Cutting Blowpipe.* The summaries of the data obtained from cutting tests made with the various blowpipes are shown in Table 2 and Figs. 8 to 15. The study of these data shows some rather interesting conditions relative to the art of metal cutting by the oxy-acetylene process, as follows:

TABLE 2 GENERAL SUMMARY OF CUTTING-TORCH DATA

BUREAU OF STANDARDS INVESTIGATION

Torch No.	Tip No.	Pressure delivered at torch handle		Length cut per hour, ft.	Volume consumption		Velocity of oxygen jet, ft. per sec.	Length cut per cubic foot of oxygen, ft.	Cost per linear foot cut, dollars.
		Oxygen, lb. per sq. in.	Acetylene, lb. per sq. in.		Oxygen, cu. ft. per hr.	Acetylene, cu. ft. per hr.			
(1)[1]	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)

TABLE 2 GENERAL SUMMARY OF CUTTING-TORCH DATA (*Cont.*)

TEST 6 (¼-in. material)

1	2	19.32	3.95	90	74	12	1975	1.45	0.025
2	1	14.79	2.98	63	37	8	922	2.17	0.025
3	1	24.70	2.98	86	53	10	1184	2.00	0.022
4[2]	....	.....	.....	.....	.....	.....	.....	.....	.....
5	2	9.53	4.97	73	50	10	752	1.82	0.025
6	2	19.21	2.98	102	77	8	1010	1.48	0.021
7	1	29.69	4.94	81	55	13	1272	1.93	0.024
8	1	24.52	5.90	98	70	20	1180	2.00	0.024
9	3	39.74	4.98	84	59	7	1502	1.61	0.022
10	1	24.68	6.97	87	57	12	1364	1.93	0.023
11	1	5.74	1.94	33	31	9	530	1.50	0.046
12	2	11.04	1.99	88	74	9	834	1.35	0.024
13	1	24.82	0.88	79	51	17	1139	2.32	0.026
14	2	28.34	1.72	109	130	8	1564	0.89	0.027

TEST 7 (2-in. material)

1	2	56.05	3.96	45	163	13	2523	0.300	0.079
2	2	42.70	2.97	46	134	8	1842	0.366	0.066
3	3	37.10	4.86	57	210	20	1814	0.300	0.078
4[2]	....	.....	.....	.....	.....	.....	.....	.....	.....
5	3	23.08	4.95	26	87	11	863	0.342	0.091
6	3	48.09	3.89	55	175	16	1805	0.346	0.070
7	2	53.70	4.93	51	154	14	2050	0.364	0.068
8	2	44.83	3.91	28	130	15	1934	0.243	0.111
9	4	58.97	5.99	44	139	9	2188	0.338	0.072
10	2	33.84	4.92	37	123	15	1580	0.342	0.083
11	2	28.95	2.90	28	104	15	1302	0.314	0.099
12	2	21.91	2.97	25	98	10	1129	0.284	0.102
13	2	44.26	0.72	39	140	25	1683	0.339	0.092
14	3	45.91	1.95	48	243	11	2979	0.207	0.099

TEST 8 (6-in. material)

1	4	74.70	4.02	29	546	25	2865	0.056	0.332
2	3	72.60	4.91	27	285	15	2260	0.100	0.206
3	4	75.83	5.74	27	517	30	2532	0.055	0.350
4[2]	....	.....	.....	.....	.....	.....	.....	.....	.....
5	4	.....	6.96	20	256	15	1824	0.083	0.253
6	5	82.20	3.91	18	479	16	2455	0.039	0.478
7	3	90.60	9.88	26	378	23	2970	0.073	0.273
8	3	69.99	3.90	27	342	16	2466	0.083	0.238
9[2]	....	.....	.....	.....	.....	.....	.....	.....	.....
10	4	72.38	3.84	24	441	21	1589	0.057	0.327
11	3	103.20	6.83	26	360	36	3365	0.080	0.280
12	4	67.39	3.63	13	241	18	2861	0.058	0.372
13	3	79.30	0.25	29	388	41	2904	0.084	0.267
14	4	70.10	3.42	27	463	14	3761	0.060	0.301

TABLE 2 GENERAL SUMMARY OF CUTTING-TORCH DATA (*Cont.*)

TEST 9 (10-in. material)

1	5	129.80	5.73	10.0	1050	44	4012	0.0100	1.776
2[2]	....	.....	.....	.....	.....	.....	.....	.....	.....
3	5	123.90	7.68	6.8	1122	33	3385	0.0064	2.725
4[2]	....	.....	.....	.....	.....	.....	.....	.....	.....
5[2]	....	.....	.....	.....	.....	.....	.....	.....	.....
6[2]	....	.....	.....	.....	.....	.....	.....	.....	.....
7	4	153.00	11.88	7.6	809	22	4350	0.0102	1.780
8	4	165.70	3.79	6.5	1157	25	4281	0.0053	2.898
9[3]	....	.....	.....	.....	.....	.....	.....	.....	.....
10	4	152.90	14.65	10.6	858	43	3085	0.0135	1.400
11	4	164.17	4.63	7.1	701	37	4565	0.0106	1.737
12	6	160.24	5.73	7.3	1216	28	4912	0.0058	2.716
13	4	117.32	0.69	9.4	816	53	2889	0.0117	1.542
14[3]	....	.....	.....	[4]	.....	.....	.....	.....	.....

[1] Numbers in parentheses are column numbers.

[2] Not submitted to cutting tests.

[3] Torch failed to cut in this test.

[4] By inadvertant error a No. 4 tip was used instead of a No. 8a in making the 10-in. cut reported on above. On a previous test with an 8a tip, Torch No. 14 made a cut through 10-in. material using an oxygen velocity of 7310 ft. per sec.; consuming 1340 cu. ft. of oxygen and 18.55 cu. ft. of acetylene per hr.; cutting at the rate of 9.23 ft. per hr. at a cost of $2.32 per linear foot of cut which is at the rate of 0.0067 ft. of cut per cubic foot of oxygen. The cut was of fair quality.

a The very great irregularity in the shape of the graphs indicates quite clearly that if the individual blowpipes have been designed according to a theory, there does not exist any generally accepted theory. In cutting identical material under standardized conditions the various blowpipes exhibit, as indicated by the data for cutting $\frac{1}{2}$-in. metal, variations in speed of cutting of 330 per cent; variations in volume of oxygen consumed per hour of 430 per cent; variations in velocity of exit of cutting-jet oxygen of 390 per cent, and variations in volume of acetylene consumed of 300 per cent.

b In general the speed of cutting increases with the volume of oxygen consumed, provided the velocity of exit of the cutting jet and the amount of acetylene consumed in the preheating flame are kept constant.

c With the same velocity of exit of cutting jet and equal oxygen consumption the speed of cutting is reduced and therefore the cost of operation increased by an increase in the amount of acetylene consumed in the preheating flames, the increased amount of acetylene requiring a larger amount of oxygen for its combustion and thus decreasing that available as an oxidizing agent in the cutting jet.

d With equal acetylene consumption and equal oxygen consumption the rate of cutting, and therefore the decrease in cost of

TABLE 3 SUMMARY OF TESTS FOR "FLASHBACK," ORDINARY CONDITIONS

BUREAU OF STANDARDS INVESTIGATION, TEST 3

Torch No.	Tip[1] No.	Pressures used, lb. per sq. in.		Action of torch in tests with			Velocity of exit, ft. per sec.	Valve wide open	
		Oxygen	Acetylene	Cold steel	Fire-brick	Wood			
(1)[2]	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	
1	6	4.00	5.00	*b, r*	.....	*FOO, b, r*	371	Oxygen	2 trials
	6	4.00	5.00	*b, v*	.....	.....	371	Oxygen	
	8	5.75	7.00	.....	.....	*v*	380	Acetylene	
2	8	8.00	8.00	*b*	*v*	*v*	364	Acetylene	
	10	10.00	10.00	*b*	.....	.....	397	Oxygen	
3	9[3]	6.50	8.00	*b*	.....	.....	479	Oxygen	
	10	7.50	9.00	.....	*F*	*FOO, F*	427	Oxygen	
4	35	12.00	5.00	*F*	*F*	*F*	366	Oxygen	2 trials
	35	12.00	5.00	*F*	*F*	*F*	366	Oxygen	
	36	14.00	6.00	*F, r*	*F*	*F*	335	Acetylene	3 trials
	36	14.00	6.00	*F, r*	*F*	*F*	335	Acetylene	
	36	14.00	6.00	*F, r*	*F*	*F*	335	Acetylene	
5	11[4]	11.00	11.00	*b*	.....	.....	450	Oxygen	
	15[5]	15.00	15.00	*b*	.....	.....	332	Acetylene	
6	7	14.00	6.00	*b*	*b*	*FAO, b*	406	Acetylene	2 trials
	7	14.00	6.00	*b*	*b*	*FAO*	406	Oxygen	
	9	18.00	6.00	*F*	*F*	*F*	375	Acetylene	3 trials
	9	18.00	6.00	*F*	*F*	*F*	375	Acetylene	
	9	18.00	6.00	.....	*F*	*FAO*	375	Acetylene	
7	7[6]	10.00	10.00	*b*	*b*	*b*	413	Oxygen	
	7	12.00	12.00	*b*	.....	*b*	462	Oxygen	
8	6	12.00	6.00	*b*	*b*	*b*	495	Acetylene	
	7[7]	14.00	8.00	*F, b*	*F*	*F*	558	Oxygen	5 trials
	7	14.00	8.00	*b*	*b*	*b*	558	Oxygen	
	7	14.00	8.00	*b*	*b*	*b*	558	Oxygen	
	7	14.00	8.00	*b*	*F*	.....	558	Oxygen	
	7	14.00	8.00	*b*	.....	.....	558	Oxygen	
9	8[8]	5.00	4.00	*b*	*b*	*b*	427	Acetylene	
	9	8.00	8.00	*b*	*b*	*F*	391	Acetylene	4 trials
	9	8.00	8.00	*b*	.....	*F*	391	Acetylene	
	9	8.00	8.00	*b*	.....	*b*	391	Acetylene	
	9	8.00	8.00	.....	.....	*b*	391	Acetylene	
10	7	8.00	6.00	*b, v*	*b, r*	*F, r*	568	Acetylene	2 trials
	7	8.00	6.00	.....	.....	*b*	568	Acetylene	
	9	14.00	14.00	*b*	.....	*F*	582	Oxygen	2 trials
	9	14.00	14.00	.....	.....	*b*	582	Oxygen	
11	5	6.00	6.00	*b*	*b*	*b, r*	430	Oxygen	
	6[9]	15.00	6.00	*b*	*b*	*b*	483	Acetylene	
	6[10]	15.00	6.00	*b*	*b*	*b*	483	Acetylene	
12	8	4.50	4.50	*b*	.....	*b*	370	Acetylene	
	9	5.50	5.50	*b*	.....	.....	335	Aceytlene	
13	10[11]	21.00	1.00	*F*	*F*	*F*	323	Oxygen	
	12	25.00	1.00	*F*	*F*	*F*	343	Oxygen	2 trials
	12	25.00	1.00	*F*	*F*	*F*	343	Oxygen	
14	8[12]	8.00	8.00	*b*	*b*	*b*	353	Acetylene	
	12[13]	12.00	12.00	*b*	.....	.....	485	Acetylene	

(For notes to Table 3 see following page)

NOTES TO TABLE 3

F = flashback.
FOO = flashback in oxygen observation tube.
FAO = flashback in acetylene observation tube.
b = backfire.
r = trouble in relighting.
v = flame blew away and valve had to be readjusted.

[1] Smaller-size tips were used for welding ¼-in. metal and larger-size for ¾-in. metal.
[2] Column numbers.
[3] Small explosions denoting attempt at backfire in tests on brick and wood.
[4] Two backfires in 6 trials on cold steel.
[5] Too hot to hold after test on wood.
[6] Head quite hot.
[7] Half of handle too hot to hold after test on wood.
[8] Tendency to flash back not sustained with brick and wood.
[9] Monel metal tip.
[10] Copper tip.
[11] Torch head very hot.
[12] No backflash after brick was warm.
[13] Tendency to backfire with wood and brick.

operation within limits, increases with the velocity of exit of the oxygen cutting jet.

e With equal acetylene consumption, increasing the velocity of exit of the oxygen jet and at the same time the oxygen consumption does not necessarily increase the speed of cutting in commensurate ratio, and may even tend toward increased cost of operation. It seems apparent that there is an economical limit to such increases and that with too high a velocity an increased oxygen volume may be wasted. This indicates that for any thickness of metal there is a limiting velocity at which complete utilization of the oxygen takes place and a limiting value to the amount of oxygen required to produce the cut.

63 As indicated above, there is a great deal of difference between the characteristics of blowpipes of different designs. One of the most interesting and noticeable differences is the fact that no make is equally proficient and economical for all thicknesses of metal. Of the six most economical cutting blowpipes on ½-in. material, two failed to cut 10-in. material and the remainder stand almost in the inverse order of rating. Further, two of the most uneconomical blowpipes on ½-in. material are two of the most economical on the heavy or 10-in. cutting. It is interesting to note that this reversal of performance in general seems to be progressive with the variation in thickness of metal cut. (See Table 2 and Figs. 12 to 15.)

64 The reason for reversal of performance, upon examination of the data available, seems to be quite clearly indicated. The average design of blowpipe is operated with the oxygen jet issuing

from the tip at too great a velocity. It will be noted that the most efficient blowpipes are those operating at the lower velocities for the metal of various thicknesses. Where there appears to be a divergence from this fact, it will be further noted that those blowpipes seeming in performance to be inconsistent with the above statement are those using an excessive amount of acetylene.

65 A portion of the data of Table 2 plotted to show the rela-

TABLE 4 SUMMARY OF TESTS FOR FLASHBACKS, SEVERE CONDITION
BUREAU OF STANDARDS INVESTIGATION, TEST 4

	Flashback caused, in minutes		Conditions prevailing after flashback	
Torch No.	First trial	Second trial	First trial	Second trial
(1) [1]	(2)	(3)	(4)	(5)
1	1.60	1.96	Tip loose	Tip loose
2	0.75	1.13		Slight leak at tip
3	0.43	0.75	Tip leaks in head when hot. No leak when cold	Same as in first trial
4	0.40	0.38		
5	0.50	0.50		Slight leak in tip
6	0.53	0.90	Had to be cooled before relighting	Tip leaked at head
7	0.86	0.72	Leak in top of tip, tip loose	Flashed back and relit, leaking around tip, which was loose
8	0.68	0.65	Tip leaked at joint	
9	0.80	0.68		
10	0.33	0.42		Small leak in bottom section of tip
11	0.80	0.92	Monel metal tip	
11J	0.38	0.58	Copper tip, tip leaked around end	
12	0.87	0.68		
13	1.07	0.85		
14	0.78	0.90		Handle too hot to hold

[1] Numbers in parentheses are column numbers.

tion between area of metal cut per cubic foot of oxygen consumed for the various thicknesses is of further interest, as it suggests that none of the blowpipes submitted to test is as economical in cutting 2-in., and possibly 10-in., metal as might be expected.

66 The law governing the relation between the area of metal cut per cubic foot of oxygen consumed for various thicknesses has not been determined during this investigation. The data indicate that

very probably a straight-line relation exists, in which case the tests would show further that there is a definite limit to the thickness (about 12 in.) which can be cut by the oxy-acetylene process.

67 On the basis of the straight-line relation it is very evident

TABLE 5 SUMMARY OF TESTS FOR GAS RATIOS
BUREAU OF STANDARDS INVESTIGATION

	Test Number						
Torch No.	1a	1b	2	5a1	5b	With gratings	
						5a1	5b
(1) [1]	(2)	(3)	(4)	(6)	(5)	(7)	(8)
1	1.13	1.16	1.14	1.19	1.08	1.04	1.01
	(1.11) [2]	(1.15)	(1.14)	(1.18)	(1.04)	(1.05)	(0.981)
2	1.12	1.07	1.08	1.04	1.06	1.04	0.992
	(1.11)	(1.06)	(1.06)	(1.04)	(1.04)	(1.02)	(0.996)
3	1.21	1.26	1.13	1.10	1.07	1.01	1.04
	(1.18)	(1.23)	(1.19)	(1.07)	(1.09)	(1.01)	(1.09)
4	1.13	1.09	1.41	1.14	1.29	.....	.....
	(1.13)	(1.09)	(1.16)	(1.15)	(1.10)	.....	.....
5	1.07	1.05	1.03	1.02	1.04	1.01	0.999
	(1.05)	(1.04)	(1.13)	(1.02)	(1.10)	(1.01)	(1.03)
6	1.10	1.11	1.14	1.12	1.17	1.06	1.08
	(1.10)	(1.11)	(1.15)	(1.12)	(1.16)	(1.03)	(1.07)
7	1.07	1.05	1.03	1.04	1.07	1.05	1.02
	(1.04)	(1.03)	(1.02)	(1.01)	(1.03)	(1.01)	(0.994)
8	1.13	1.12	1.18	1.27	1.43	1.05	1.01
	(1.12)	(1.12)	(1.16)	(1.28)	(1.38)	(1.03)	(0.986)
9	1.15	1.18	1.19	1.14	1.14	1.43	1.19
	(1.18)	(1.24)	(1.17)	(1.14)	(1.15)	(1.39)	(1.15)
10	1.12	1.15	1.19	1.10	1.04	1.03	1.01
	(1.11)	(1.13)	(1.06)	(1.09)	(1.05)	(0.989)	(0.992)
11	1.19	1.20	1.26	1.27	1.09	1.21	1.04
	(1.19)	(1.21)	(1.23)	(1.28)	(1.08)	(1.19)	(1.05)
12	1.21	1.21	1.13	1.02	1.04	1.02	1.02
	(1.17)	(1.20)	(1.11)	(1.01)	(1.02)	(0.998)	(0.979)
13	1.02	1.13	1.09	1.09	1.07	1.02	1.00
	(1.04)	(1.12)	(1.09)	(1.08)	(1.06)	(1.01)	(1.00)
14	1.07	1.09	1.08	1.08	1.10	0.999	0.956
	(1.06)	(1.08)	(1.07)	(1.08)	(1.10)	(0.992)	(0.964)

[1] Numbers (1) to (8) across page are column numbers.
[2] Values in parentheses are computed from flowmeter data.

that none of the blowpipes tested was cutting 2-in. material economically. It would seem that the reason for this non-economy of operation is probably due to two things: First, the use of an oxygen cutting jet issuing at too great a velocity. If a velocity of 2200 ft. per sec. is sufficient for cutting 6-in. material, it would appear that

TABLE 6 SUMMARY OF RESULTS OF TENSILE AND BEND TESTS OF OXY-ACETYLENE WELDS

BUREAU OF STANDARDS INVESTIGATION, TESTS 1*a*, 1*b*, 1*c*, 1*d*, 2

	Test Number									
	1*a*		1*b*		1*c*		1*d*		2	
Torch No.	Ult. T. S.,[1] 1000 lb. per sq. in.	Included angle of bend[2], deg.	Ult. T. S., 1000 lb. per sq. in.	Included angle of bend, deg.	Ult. T. S., 1000 lb. per sq. in.	Included angle of bend, deg.	Ult. T. S., 1000 lb. per sq. in.	Included angle of bend, deg.	Ult. T. S., 1000 lb. per sq. in.	Included angle of bend, deg.
(1)[3]	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	10)	(11)
1	39.3	116	44.4	115	33.1	44	42.1	77	43.6	51
	44.9	121	47.4	95	.....	.....	.....	.....	.....	.....
2	41.2	29	39.4	39	41.2	67	45.8	58	43.7	69
	44.1	56	43.9	67	.....	.....	.....	.....	.....	.....
3	31.9	50	27.9	28	43.0[4]	94[4]	35.3	37	33.2	30
	32.1	36	35.9	20	38.9[5]	62[5]	.....	.....	.....	.....
4	36.2	62	41.4	95	40.6	61	39.0	107	53.5	117
	43.0	41	36.7	73	.....	.....	.....	.....	.....	.....
5	43.0	72	31.1	63	43.6	34	32.1	18	33.0	52
	46.3	25	37.7	63	42.9[5]	42[5]	.....	.....	.....	.....
6	40.6	72	45.8	59	34.4	16	44.3	56	48.6	44
	43.7	69	42.3	65	.....	.....	.....	.....	.....	.....
7	44.2	33	37.3	56	36.1	73	36.3	53	48.2	49
	46.0	77	40.8	65	.....	.....	.....	.....	.....	.....
8	43.4	43	39.8	64	32.9	20	39.5	80	41.4	74
	43.8	70	42.6	60	.....	.....	.....	.....	.....	.....
9	.....	.....	34.9	38	42.0	67	44.1	71	44.9	57
	46.5	62	45.2	39	.....	.....	.....	.....	.....	.....
10	41.9	54	35.1	26	47.7[4]	89[4]	45.6	72	39.4	23
	43.1	59	39.6	51	.....	.....	.....	.....	.....	.....
11	43.8	54	38.7	66	37.9	36	38.4	96	45.1	39
	27.0	51	29.2	57	32.8	38	.....	.....	.....	.....
12	40.4	43	39.0	54	42.3	60	49.4	94	45.8	62
	40.2	32	39.5	56	.....	.....	.....	.....	.....	.....
13	39.3	51	31.0	83	40.1	61	34.4	82	48.3	58
	52.1	61	41.6	24	.....	.....	.....	.....	.....	.....
14	40.4	63	45.6	107	39.4	52	42.4	65	38.1	38
	41.0	49	46.5	66	.....	.....	.....	.....	.....	.....
Average:										
1st plate	40.4	57	38.0	64	38.6	49	40.6	69	43.3	54
2d plate	42.4	58	40.6	57	32.8	38	.....	.....	.....	.....
Gen. Avg.[6]	41.4	57	39.3	60	35.7	44	40.6	69	43.3	54

Average ultimate tensile strength of all welded $\frac{1}{4}$-in. plates..... 39.2 lb. per sq. in.
Average ultimate tensile strength of unwelded $\frac{1}{4}$-in. plates..... 54.9 lb. per sq. in.
Efficiency of welds in $\frac{1}{4}$-in. plate..... 71.4 per cent.
Tests 1*a* to 1*d*, inclusive, were welds made with $\frac{1}{4}$-in. plates; Test 2 with $\frac{3}{4}$-in. plate.

(For notes to Table 6 see following page)

NOTES TO TABLE 6

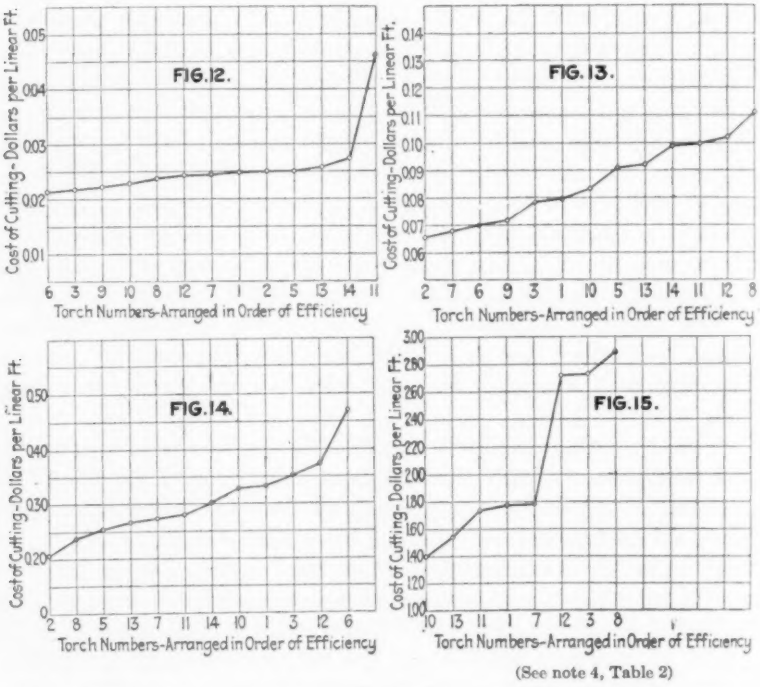
[1] Ultimate tensile strength in thousands of pounds per square inch.
[2] Included angle of bend, cold-bend test. Bottom of V in compression. Pin diameter equal to thickness of metal. Included angle for unwelded plate = 180 deg.
[3] Numbers in parentheses are column numbers.
[4] Weld made at — 50 per cent pressure instead of at + 50 per cent pressure.
[5] Weld made at normal pressure instead of at + 50 per cent pressure.
[6] Welds referred to in [4] and [5] not included in the general average.

a velocity of 1800 ft. per sec. is excessive for 2-in. material. Secondly, in the blowpipes tested the preheating flames in most cases were too large for the work required. In fact, in a number of instances there was practically very little difference between the size of the preheating flame used for the 2-in. plate and that used for the 6-in. material. For the thinner material the large-sized preheating flame would tend toward non-economy in operation, inasmuch as it would preheat a relatively large surface with the resulting tendency for the oxygen to oxidize the entire body of metal thus preheated, giving probably a wider kerf than necessary and thus using the oxygen supply non-advantageously. This conclusion is further substantiated by the fact that in cutting 2-in. material practically none of the slags showed metallic characteristics, but was on the other hand fully oxidized metal.

68 A further study of the data indicated also that possibly the maximum economy has not been fully secured for the 10-in. material. The cuts for this thickness also indicate that the maximum limit for economical cutting was being approached, as shown by Figs. 18 to 20. As an interesting side light on this, Fig. 22 shows an attempted cut for maximum thickness made with the blowpipe that exhibited maximum efficiency in heavy cutting. The cut penetrated about 12 in. into a 16-in. block, the bottom of the cut being practically at right angles to the direction of cut.

69 This immediately brings up the question as to why the limit of depth of cut for an oxy-acetylene blowpipe should apparently be about 12 in. It has been known that in heavy cutting the cleaner cuts and greater depths were secured with the oxy-hydrogen blowpipe (see Fig. 21). The reason generally given for this difference is that the oxy-hydrogen preheating flame is longer and therefore secures better preheat penetration within the cut. It would seem, however, that a possibly more probable explanation of the difference lies in the resulting products of combustion and that this is also a possible explanation of the limitation of the oxy-acetylene blowpipe in heavy cutting. Besides the iron oxide, water

vapor and carbon dioxide are formed during the cutting process. With the use of acetylene instead of hydrogen a greater proportion of carbon dioxide is formed. This gas sinking to, and being partially trapped in, the lower portion of a cut and being a non-supporter of combustion, causes a slackening of the oxidation process and in deep cuts it may be present in sufficient volume to actually arrest



(See note 4, Table 2)

FIGS. 12 TO 15 COST OF CUTTING METAL, PER LINEAR FOOT, WITH OXY-ACETYLENE BLOWPIPES

(Fig. 12, for ¼-in. metal; Fig. 13, for 2-in. metal; Fig. 14, for 6-in. metal; Fig. 15, for 10-in. metal)

the oxidation process. This suggestion would seem to be verified by the experience that in deep cutting a large increase in the kinetic energy of the issuing cutting jet produced by increasing the pressure at which the gas is used will often secure a cut where the blowpipe would otherwise not have been enabled to do so. The increased pressure and resulting energy insure that the products of combustion are blown out of the way. Under such conditions increased depths of cuts may be secured but not necessarily economically.

70 Unfortunately these questions have not been definitely settled by these tests and the above must therefore be looked upon simply as indications and suggestions for further work toward securing increased economy in cutting.

71 In questions of economy of operation the volume of gas used plays an important part. Where an excessive amount of acetylene is used, acetylene being the more expensive gas, the economy of operation is affected adversely. This question of acetylene consumption is one that would seem to deserve attention. The size of the preheating flame is a direct index of the acetylene consumption. As indicated above, in a great many of the blowpipes tested it was found necessary to reduce the acetylene pressure under that specified by the manufacturer (thereby reducing the size of the flame or acetylene consumption) in order to secure the flame size that did not melt too much of the top edge of the cut and reunite the pieces cut apart by the flow of this molten metal back into the kerf. In some instances with the specified preheating flame the pieces were practically welded together again along the top edge, and in others the top edge was badly beveled.

72 Neither of the above conditions, developed by excessive-sized preheating flames, is desirable. Even though not welded together, a badly beaded top edge (Fig. 17*H*) or a badly beveled top edge would increase the cost of following manufacturing operations in most cases. The feeling of the blowpipe manufacturers on this matter seems to be that they must have excessive preheating flames to take care of heavy scale, slag, etc., and to facilitate rapid cutting, especially at the start of a cutting operation. The experience of this investigation as developed on cutting does not indicate that a cut can be started any quicker with an excessive flame than with one of proper size. In cases where slag or heavy scale are met with it is doubted whether burning slag and scale off at an oxygen cost of $1.50 to $10.00 an hour (according to thickness of material cut) can economically compete with unskilled hand labor (chisel and hammer) at 75 cents per hour.

73 The quality of cut secured is quite important. Figs. 16 and 17 are included to show what may be expected from a properly operating cutting blowpipe. Figs. 18 to 20, of attempted cuts on 10-in. material, in this case armor plate, speak for themselves and they may well be taken to indicate that none of the blowpipes tested is suitable for working to close limits on cuts of such depths. For comparison an oxy-hydrogen cut on the same block is shown in Fig. 21.

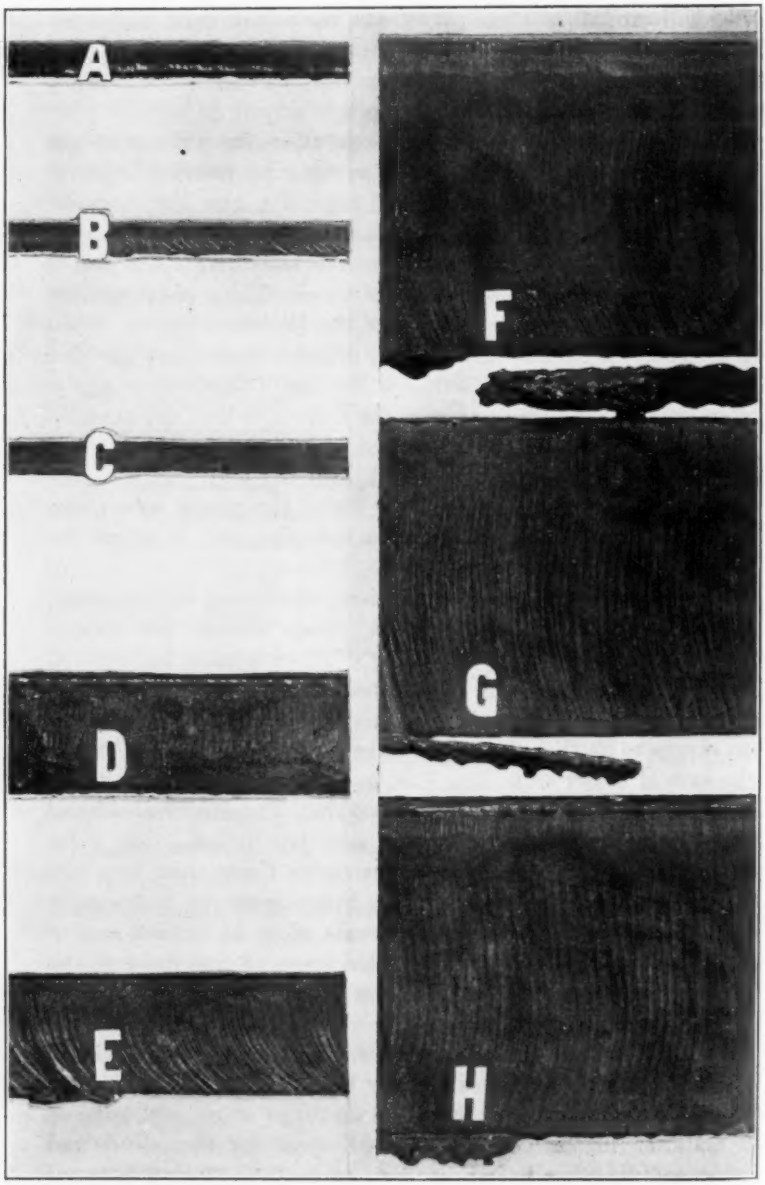


Fig. 16 Fig. 17

FIGS. 16 AND 17 CUTS FROM PROPERLY OPERATING BLOWPIPES

74 On the other hand, there is splendid evidence that some blowpipes can do especially nice work when properly designed for the thickness being cut, in this case 6 in. (Fig. 17*G*). A sectional view through this cut is shown in Fig. 23. Fig. 16, *B* to *E*, inclusive, show typical cuts of 2-in. and ½-in. metal, and Fig. 16*A* is further evidence that there is such a thing as using too much oxygen in executing a cut.

75 Besides the top edge and face of a cut, the bottom edge is of interest as its appearance is quite closely connected with the type of slag formed. A completely oxidized slag has a greater tendency

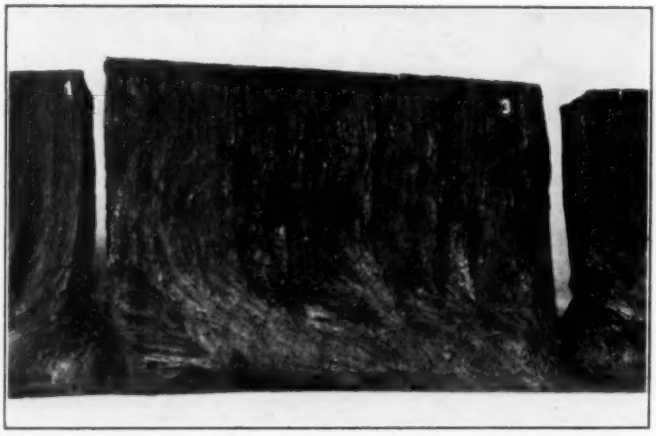


FIG. 18 ATTEMPTED CUTS ON 10-IN. MATERIAL

to cling to the under side, and if excess oxygen is present there is considerable undercutting which tends to increase the amount of slag and to make it adhere more firmly, with the result that a rough bottom edge is produced. By the proper proportioning of the oxygen volume it is apparent that a slightly metallic slag can be secured, the metallic element being due to the fact that in the presence of an insufficient oxygen volume some of the metal is blown out without being oxidized. A slightly metallic slag clings quite firmly to the bottom edge of the cut, but apparently possesses an advantage in that it may be removed in long pieces as shown in Fig. 17*G* and leaves behind it, as seen, a very smooth, sharp under edge. The section of the 6-in. cut shown in Fig. 23 had a slightly metallic slag.

76 In the use of the oxy-acetylene cutting blowpipe with especial reference to manufacturing operations the question arises as to what effect the cutting action has upon the metal body.

77 Macroscopic examination of sections through cuts ½-in., 2-in. and 6-in. metal, Fig. 23, show the depth to which the metal has been affected by the cutting process. Microscopic examination of these sections indicated that possibly the affected layer had been carbonized. To determine this, a section of the affected layer and of the interior portion of the specimen *C*, Fig. 23, were annealed and reëxamined. The results obtained are shown in Fig. 24.

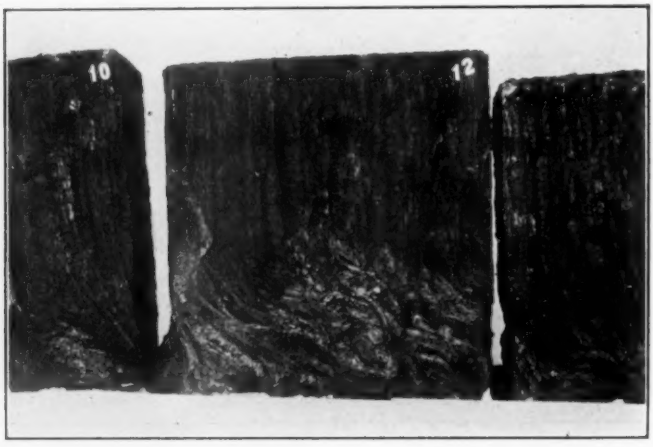


FIG. 19 ATTEMPTED CUTS ON 10-IN. MATERIAL

78 From this study it may be concluded that the appearance of the surface layer of the steel directly after cutting with the oxy-acetylene flame is due to the chilling action of the large mass of unheated metal upon the hot surface after the flame has been removed. Such an effect would, of course, be more noticeable in a steel of higher carbon content than in one which is lower in this element. This of course accounts for the hard surface metal that is frequently encountered in working up metal cut with the oxy-acetylene blowpipe.

79 *The Welding Blowpipe.* It is universally accepted that outside of the mechanical features of design that affect weight, balance and convenience of operation, the prime essentials of a strictly satisfactory piece of apparatus are:



FIG. 20 ATTEMPTED CUTS ON 10-IN. MATERIAL

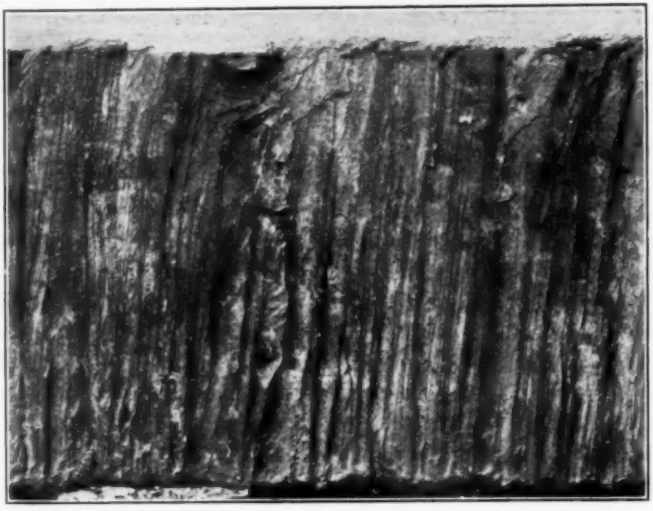


FIG. 21 OXY-HYDROGEN CUT ON 10-IN. MATERIAL

a Safety under all operating conditions

b Freedom from the so-called phenomenon of "flashback" or sustained backfire

c The quality of maintaining under all operating conditions a welding flame that is neither oxidizing nor carbonizing, one technically known as a "neutral flame," which in the process of combustion consumes, as nearly as possible, equal volumes of oxygen and acetylene; that is, maintains as nearly as possible the theoretical gas-volume ratio of unity.

80 The tests of this investigation were decided upon with the

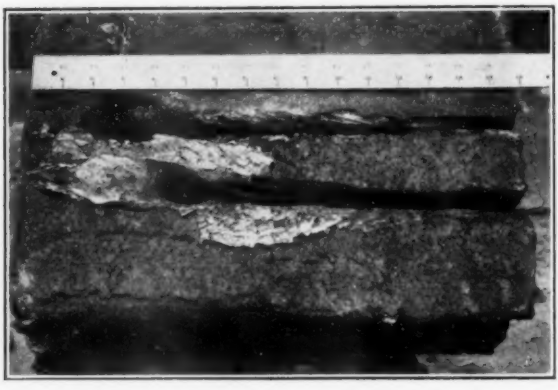


FIG. 22 ATTEMPTED CUT FOR MAXIMUM THICKNESS

idea of furnishing data that would enable blowpipes to be compared with respect to these essentials.

81 A study of the data obtained at the completion of the prescribed tests showed so many apparent inconsistencies that it was evident that there was a governing factor that was not understood, and that was, so far as test data were available, not in evidence. Irrespective of the fact that particular attention had been given to insure identical working conditions and gas-pressure control, and that especial care was taken to secure exceedingly competent and unbiased operators, the results obtained from the welding tests seemed extremely unsatisfactory. Gas ratios obtained during actual welding operations were extremely high. Those obtained when the blowpipe was burning freely in air were also higher than was to be expected. In tests for flashbacks there seemed to be a difference

in the ease with which they could be developed in blowpipes of different manufacture, but there appeared to be no criterion that would enable one to say just why such phenomena could be caused more easily in some pieces of apparatus than in others, or why with some pieces of apparatus flashbacks could be produced at times quite readily and at other times with difficulty. Finally the general quality of the welds produced during test, although executed with the greatest care and shown by tensile tests to be of a higher strength

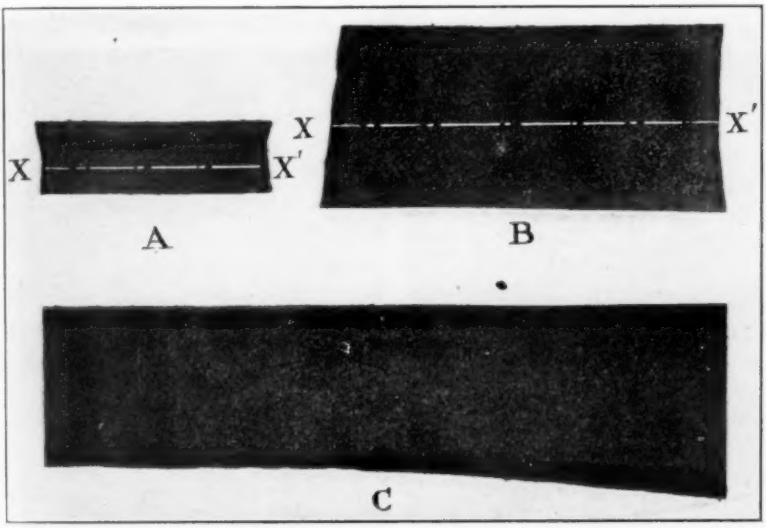


FIG. 23 EFFECT OF CUTTING ACTION ON METAL BODY

than is generally secured in most welding shops, was far from satisfactory.

82 It was evident that further information was essential if a satisfactory analysis of welding-blowpipe performance was to be made. The greatest discrepancy from what was expected appeared to be in the high gas ratios obtained and the first attempt to answer the problem was made by a study of gas-ratio phenomena. This seemed particularly desirable as it has been very firmly held by almost all authorities that good welding cannot be done with blowpipes having a high gas ratio.

83 An extensive series of supplementary gas-ratio tests was

accordingly carried out. As a result of these tests a phenomenon of exceeding interest was developed, and gas ratios approaching very nearly the theoretical value obtained. The value of these tests will be considered later under the subject of gas ratio. The results obtained accounted partly for the quality of the welding work secured. In some respects though they added increased confusion by lack of consistency.

84 In further study of the data secured by the prescribed tests, it was noticed that blowpipes that seemed especially susceptible to

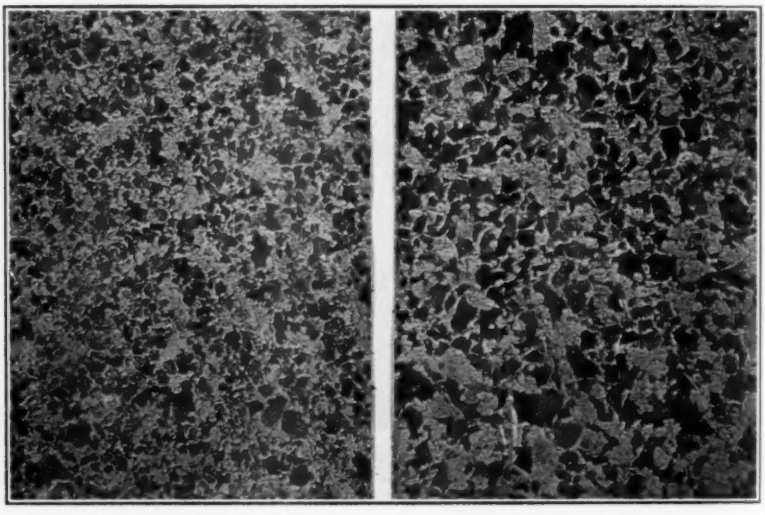


FIG. 24 EFFECT OF CUTTING ACTION ON METAL BODY

flashback were those in which the oxygen was delivered to the blowpipe at a pressure very much in excess of that at which the acetylene was delivered. It was further noticed that even among such blowpipes inconsistencies appeared. Critical examination of tip design in these blowpipes suggested a possible explanation. On this basis another series of supplementary tests was made, from the results of which data were secured that clearly explain the cause of flashback phenomena.

85 The logical continuance of the theory evolved for flashback leads directly to the question of safety in operation and correct

gas ratio and will explain, in large part, the reason for so large a proportion of oxy-acetylene welds being of inferior grade.

86 The essential qualifications for a satisfactory welding blowpipe as enumerated above are therefore very intimately connected with the conditions governing flashback. It seems desirable, then, to begin the discussion of the results obtained in the tests of the investigation by the critical analysis of the conditions conducive to the development of flashbacks in welding blowpipes.

87 Before taking up this discussion it will be of value to define the term "flashback" or, as it is sometimes called, "sustained backfire." It was very evident throughout the investigational tests that the representatives of the manufacturers were not in agreement as to what constitutes a flashback. To some the terms "flashback" and "backfiring" were synonymous, others distinguished between the blowpipe action representing these terms, and at least one other manufacturer attempted to distinguish blowpipe behavior by incorporating a term "preignition." There is therefore an apparent confusion of ideas.

88 For the purpose of this discussion the following definitions have been accepted as representative of blowpipe behavior under certain operating conditions:

(*a*) *Backfire or Preignition.* Momentary retrogression of the blowpipe flame into the blowpipe tip which may relight immediately upon withdrawing the blowpipe away from the test piece or necessitate the reignition of the gases by means of a lighter. In this latter condition the flame will ignite properly without manipulation of the blowpipe-handle valves.

(*b*) *Flashback or Sustained Backfire.* The retrogression of the burning flame back into the mixing chamber accompanied by the well-known marked hissing or squealing sound and the characteristic smoky, sharp-pointed flame of small volume necessitating the immediate cutting off of the gas supply by means of the blowpipe-handle valves to prevent severe heating and possible destruction of the blowpipe head, and generally necessitating cooling the head before the blowpipe can be reignited.

89 The desirability of securing a non-flashback blowpipe has generally been considered to be due to its freedom from inconvenience and loss of time during welding operations and to its safety. Probably nothing is more detrimental to the poise of an operator

and his desire to perform excellent work than an instrument that is constantly giving trouble. But the principal detriment in a blowpipe that flashes back, other than that of loss of time during welding operations, is in the possibility that a flashback may propagate itself back into the gas lines and in so doing possibly become responsible for a violent explosion with its accompanying dangers.

90 Blowpipe designers have devoted considerable time and money to the problem of minimizing or possibly obviating this latter danger and have devised a number of blowpipe designs that were expected to eliminate flashbacks. A study of a number of letters patent indicates quite clearly that there are nearly as many theories concerning the cause of flashbacks as there are patented designs. Necessarily considerable ingenuity has been shown in the attempts to accomplish the desired results. Notwithstanding the fact that some of the more recently designed blowpipes lay claim to being safe and non-flashback, still, as will be shown presently, none of those submitted to the tests of this investigation have proved free from flashback under ordinary flashback-test procedure; nor may a great many of them be considered entirely safe pieces of apparatus. The fact is that the theory back of the causes of flashback does not appear to have been fully comprehended and therefore naturally not correctly applied. Neither do any of the treatises or published articles dealing with oxy-acetylene blowpipe design or the autogenous welding process give any statements that indicate that flashback phenomena have been understood. It is essential, then, that an explanation of the phenomena be offered, especially, as indicated above, as they are intimately connected with the question of sound welds. The explanation offered is based on some very old principles.

91 Sir Humphry Davy, in the course of the researches that led to the invention of the miner's safety lamp, among other things showed that there were certain governing principles relative to the inflammability of gaseous mixtures. He pointed out that there was a relation between the richness of the gaseous mixture and the rate of propagation of flame through it, and that such gaseous mixtures could either be enriched to the point where they ceased to be inflammable, or that they could be diluted to the point where they would cease to be inflammable. It was also shown that the points where inflammability ceased were very sharply defined, but that the limits could be increased by increased temperature or pressure; and in fact that at these "critical proportion mixtures," at dilution and concentration, a very slight temperature rise will produce an inflammable

mixture. It was also shown that the maximum velocity of flame propagation occurred with the gaseous mixture so proportioned that, at the completion of the combustion process, stable compounds were formed. Finally it was shown that if a gaseous mixture is ejected through a tube into the air at a velocity greater than the velocity of flame propagation for that mixture, the mixture may be ignited and will burn at the exit opening of the tube without striking back into the interior of the tube. These fundamental principles were later proved to be correct in every particular by Bunsen, and since then by a great number of other investigators.

92 The chemistry of the combustion process of a gaseous acetylene-oxygen mixture shows that the complete oxidization of one volume of acetylene requires two and one-half volumes of oxygen. Experience with the design and operation of oxy-acetylene blowpipes has demonstrated the fact, however, that in reality it is only necessary to project, with the blowpipe, equal volumes of oxygen and acetylene, chemistry showing that it is a two-phased process, the second phase of which can be completed by the absorption of the remaining required oxygen from the atmosphere surrounding the flame.

93 The above shows that the design of the oxy-acetylene welding blowpipe is based upon the fact that equal volumes of acetylene and oxygen are to be mixed and ejected through the tube of the welding blowpipe tip at a velocity at least equal to or in excess of the velocity of flame propagation for that mixture. Under such conditions the flame will burn steadily and freely at the exit end of the tube, gathering enough oxygen from the surrounding air to complete the oxidization of the components of the acetylene gas.

94 A long experience has seemed to show, nevertheless, that the fulfilment of the above conditions was not sufficient in itself to maintain always the flame at the tube or tip end. It was found that if a blowpipe tip is brought too near the surface of the weld, is accidentally dipped into the molten metal of the weld, has the aperture partly blocked by flying slag or is badly overheated, the flame will snap back into the tube, causing a so-called backfire, and may even propagate itself still further backward until it burns at the mixing chamber, thus causing what is known as the flashback phenomenon. It is due to the fact that this detrimental phenomenon could be so readily produced that so much energy and money have been expended in the attempt to eliminate it.

95 Outside of the flashback produced through the severe over-

heating of the head and tip of a blowpipe, it is apparent that all the other causes mentioned above seem to depend for their action upon a partial or complete blockage of the gas flow. It will be readily perceived that any sort of obstruction that checks the volume flow of the gases also increases proportionally the velocity of exit of the gaseous mixture at the tip end. Such being true, it would seem that, with the velocity of exit increased above the velocity of flame propagation, the flame could not strike back into the tube—in other words, backfire. Yet in spite of this backfires are caused, and it is known that with some blowpipes a backfire will immediately become a flashback, while in others it causes the extinguishing of the flame or, if the blowpipe is withdrawn from the work, it will be blown forward again to the exit of the tip or tube. Why should some blowpipes be so much more susceptible to the development of flashback than others?

96 The lack of sufficient velocity of exit of the gaseous mixture has been accredited as the principal cause for flashback in some of the latest-designed and most successful non-flashback blowpipes. That there is some reason of far deeper significance than this is evident from a study of Tables 3 and 4. It is true that an obstruction to the flow will cause flashbacks, but such obstruction is in reality only a contributing cause.

97 Partial obstruction of the tip opening by slag, etc., or the confinement of the expanding products of combustion through bringing the tip end too near to the metal surface, causes the development of a back pressure. This back pressure may be readily observed by watching a pressure gage as a lighted blowpipe tip is alternately brought in contact with or removed from a plane surface.

98 This back pressure has a very detrimental effect, in that it chokes off the flow of the gas that is admitted under the lower pressure. Results of tests in which the amount of obstruction of the exit opening was under control show the development of this back pressure and the restriction of the flow of the gas admitted under lower pressure.

99 If, then, the blowpipe design is such that the back pressure tends to check the oxygen supply, an increased richness (in acetylene) of the gaseous mixture occurs. The gaseous mixture in the blowpipe will therefore have a lower flame-propagation velocity, due to the fact that the increased richness is tending toward the critical-proportion mixture at concentration. The velocity of flame propagation is therefore decreasing while the velocity of exit has been increased

by the partial closure of the orifice. There is a resulting tendency, therefore, for the flame to be maintained at the tip end. If under the same conditions the obstruction is sufficiently severe the gaseous mixture within the blowpipe passes the critical-proportion mixture at concentration, the mixture becomes non-inflammable, and the flame is extinguished. With blowpipes furnishing the acetylene under a higher pressure, it is therefore evident that the tendency to produce flashbacks is minimized, if not entirely suppressed.

100 On the other hand, if the blowpipe design is such that the back pressure tends to check the acetylene supply, a mixture leaner in acetylene is produced. Such a mixture is approaching the two and a half volumes of oxygen to one volume of acetylene mixture at which the amount of oxygen is in sufficient quantity to form complete stable compounds as a result of the combustion process.

101 This will be recognized, in the light of Davy's discoveries, as the mixture at which maximum velocity of flame propagation exists. A back pressure, then, that increases the oxygen content above that of the one-to-one volume ratio furnished in blowpipe design, is producing a gaseous mixture that possesses a higher flame-propagation velocity than that possessed by the one-to-one mixture.

102 If the increased exit velocity caused by the obstruction to the gas flow compares favorably with the increased velocity of flame propagation, the ignited mixture would continue to burn at the tip. In all blowpipes of this type which have been examined this condition is not fulfilled, with the result that a flashback always occurs. As experimental evidence of the correctness of this statement the data of Fig. 25 are presented. These were obtained with a blowpipe mounted so that the amount of the obstruction to gas flow could be controlled. When the oxygen content of the gas mixture had been increased to the amount indicated by the diagram, the flame snapped back of its own accord to a true flashback.

103 Having snapped back to the mixing chamber the back pressure will be destroyed, and the flow becoming normal again, the flame burns there in the volumes permitted by the blowpipe construction, with insufficient oxygen to cause complete oxidization, and the acetylene gas is broken up into its components, one of which unites with the available oxygen to form water vapor, leaving carbon deposited on the walls of the tip tube and mixing chamber. This explains not only the heavy carbon deposit that is always left as the result of a flashback, but the reason why it is relatively so moist.

104 It is of course possible that if the oxygen content of the

gaseous mixture could be increased beyond the proportions for maximum velocity of flame propagation, that is, beyond the $2\frac{1}{2}$:1 volume ratio, the velocity of flame propagation might be reduced sufficiently to compare favorably with the exit velocity. In the light of the above, however, it is evident that this condition cannot be produced—nor would it be desirable, as the flame would be extremely oxidizing.

105 At how great a disadvantage the high-pressure oxygen-blowpipe designs are as compared with those operating with acetylene delivered at excess pressure, is very apparent from another consideration. Eitner[1] has shown that the lower inflammable limit by percentage volume of combustible gas in a mixture of acetylene and

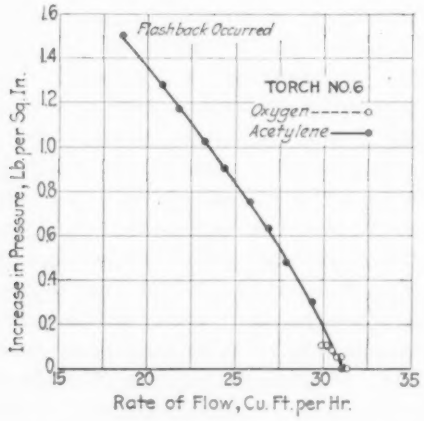


FIG. 25 REDUCTION OF FLOW DUE TO BACK PRESSURE

air is about 3.35 per cent and that the upper limit is about 52.3 per cent. There is no doubt that with a more active gas such as oxygen the lower limit will be at least the same, and possibly the upper limit will be found to be slightly raised. Bearing in mind that normally a 50 per cent mixture is being passed to the mixing chamber, it is readily seen that the critical-proportion mixture at concentration is reached very soon, while the critical-proportion mixture at dilution is not reached until almost the entire combustible-gas supply has been shut off.

[1] L. A. Groth, Welding and Cutting Materials by Aid of Gases or Electricity (1909), Archibald Constable & Co., Ltd., London, p. 251.

See also G. A. Burrell and G. G. Oberfell, The Explosibility of Acetylene, Bureau of Mines Technologic Paper No. 112, p. 5 (1915).

106 It has been known for some time that an operator could extinguish a flashback in a high-pressure blowpipe by suddenly increasing the oxygen pressure or by crimping the acetylene supply line. The reasons are now very apparent. In the former case the oxygen volume ratio is greatly increased by the accelerated checking of the acetylene supply until the volume ratio is diluted beyond the maximum explosive mixture, the flame-propagation velocity is thereby reduced and at the same time the increased pressure of the oxygen expels the mixture at increased velocity. The flame may therefore be pushed to the outlet opening of the tip. In the latter case, crimping the hose line, the combustible gas is shut off and the flame may go out from the gas mixture having reached the critical-proportion mixture at dilution, but in this case, on account of the low percentage mixture at which this occurs, it may as well be caused through the entire combustible gas supply having been cut off.

107 Having explained the conditions necessary for, and the action of, flashback phenomena, it is of interest to determine why some blowpipes flash back so much more readily than others of similar design; as for example those in which the oxygen pressure is higher than that of the acetylene. The rapidity of development of the condition for producing a flashback depends entirely upon the rapidity with which the back pressure is set up. That there is a difference in the rate at which the back pressure is developed was evident, tests showing that those blowpipes which were more susceptible to flashback from obstruction of gas flow were those in which the effects of the back pressure were developed most rapidly. A blowpipe operating on the injector principle or one in which the oxygen pressure is very much in excess of the acetylene pressure will therefore be far more susceptible to flashback phenomena.

108 An investigation of the actual pressures specified by the manufacturers as well as a study of certain letters patent indicates that a few of them at least, in their experimental work, have become cognizant of the effect a gas mixture rare in acetylene has upon the development of a flashback. To overcome flashback troubles several of the recent designs are based upon furnishing acetylene under a higher pressure, spoken of in the letters patent as higher velocity, than that used for the oxygen. Reference to Tables 4 and 5 shows that flashbacks can be caused in apparatus of that type. The question may rightly be asked as to why flashbacks can then be developed in such blowpipes.

109 A critical study of the design of blowpipe tips shows

further that the way in which one stream of gas is admitted into another has a decided effect upon conditions resulting from the development of a back pressure.

110 A back pressure caused by tube obstruction pushes back into the tube in the most direct line. (Fig. 26.) An acetylene-gas passage entering the oxygen tube at right angles has its gas flow cut off very quickly, especially when the oxygen pressure is much higher than the acetylene pressure and the point of admission of the oxygen is, as it is in most of the tip designs, beyond the point of admission of the acetylene.

111 In this respect a gas passage entering the oxygen passage at a sharper angle is more advantageous, but under certain conditions this construction may not produce results that are any more desirable. If the velocity of the oxygen jet is relatively high and the acetylene-gas ports are placed at an angle to the tube of the oxygen jet, an aspirator or injector action is developed relative to the acetylene. The higher the velocity of the oxygen the greater is the aspirating effect on the gas tube. This aspirating effect is in reality an attempt to produce a partial vacuum within the acetylene tube. If, then, while operating under such conditions a sudden obstruction of the exit passage occurs with its accompanying back-pressure development, the sudden release of the aspirating action causes a collapse of partial vacuum and the gas-passage tube is therefore momentarily choked or at least throttled by the infiltration of oxygen. This action instantly produces the leaner mixture and conditions necessary for the development of a flashback.

112 But there are also two other fundamental physical laws which apparently have not been given proper consideration in the designs of blowpipes, and which will probably account in a far larger measure for the development of flashback conditions. These laws refer to:

a The variation of velocity of a fluid in a pipe of varying size

b The diminution of pressure in a throat.

113 The first law recognizes the fact that the mass discharge of a fluid at all sections is a constant; hence for a pipe of varying size as shown, the following condition holds when the friction loss of pressure is neglected:

$$\text{If } a > a_1 \text{ then } av < a_1 v_1$$

where a and a_1 = areas of gas passage

v and v_1 = velocities of gas.

114 This indicates that at various points along a tube the velocities of the fluid increase as the section areas decrease.

115 The second law is based on the fact that, neglecting friction and eddy losses, at any two sections of such a tube the total energy per unit mass will be the same, i.e., an increased velocity produces a decreased pressure.

116 The relation for a gas is given by the equation—

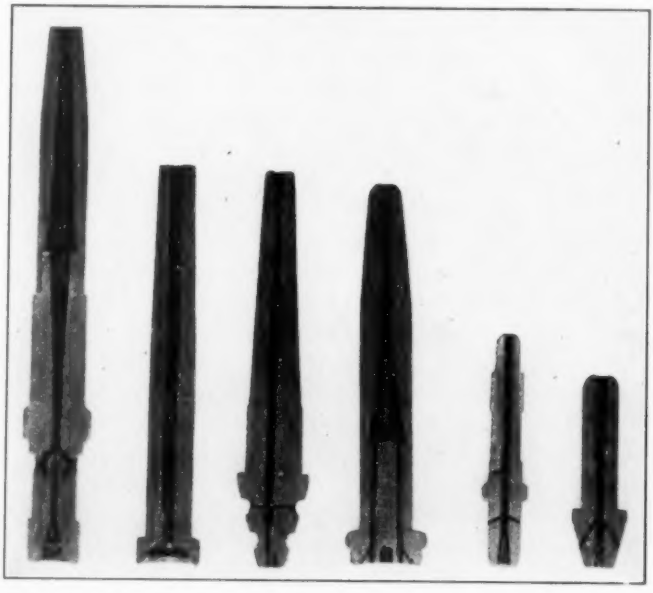


FIG. 26 CROSS-SECTIONS OF BLOWPIPES

$$v_1^2 - v^2 = 2\,TC_p\left[1 - \left[\frac{p_1}{p}\right]^{\frac{C_p - C_v}{C_v}}\right]$$

where p and p_1 = pressures
v and v_1 = velocities
d = density
T = absolute temperature
C_p = specific heat at constant pressure
C_v = specific heat at constant volume.

117 The decrease of pressure with increase of velocity is perhaps more clearly shown by the approximate formula—

$$p + \tfrac{1}{2}\, dv^2 = p_1 + \tfrac{1}{2}\, dv_1^2$$

which neglects the expansion of the gas.

118 From these two laws it is evident that if the area of a tube is contracted with the idea of securing increased velocity, the pressure in the contracted area is reduced.

119 Reference to Fig. 26 shows that in practically all designs the contracted throat is used in the acetylene passages. It will be readily realized, then, that in even those blowpipes which are supposed to operate with acetylene delivered at excess pressure there exists in the fundamentals of their design an inherent tendency for the choking off of the acetylene supply and the development of the leaner flashback mixture at the very first indication of back pressure.

120 The inherent defects in the present designs show plainly, then, why it is also possible to produce a flashback almost at will in the blowpipes that are designed on the excess-acetylene-pressure basis. It is evident, therefore, that none of the blowpipes designed for use with one gas delivered at a higher pressure than that used for the other is really free from flashback troubles caused by obstructions to the gas flow.

121 There is another important consideration to be discussed in respect to blowpipes using one gas at higher pressure, and that is their safety during operation.

122 The foregoing discussion shows that an obstruction to the gas flow issuing from the tip sets up a decided and effective back pressure. If the obstruction continues for a sufficient interval of time and is severe enough, it is readily perceived that the back pressure thus set up will be sufficient not only to check the flow of the gas admitted under the lower pressure, but also to push it actually back into its supply tube and line. The return of one gas into the supply line of the other instantly forms back of the mixing chamber an explosive gas mixture. Reference to the exceedingly low limit of the critical mixture at dilution as indicated by Eitner's investigations shows how rapidly a dangerous mixture can be formed by a blowpipe using acetylene under excess pressure. A blowpipe using the oxygen under the excess pressure is about equally as dangerous. It is simply a question of the time interval the back pressure has available to accomplish its results.

123 The degree of danger from this cause is relatively proportional also to the value of the higher pressure used. A blowpipe

operating with one gas at a pressure greatly in excess of the other can force a return mixture just so much more quickly than a blowpipe operating with less difference in pressures. Also it is readily perceived that with one pressure greatly in excess of another a larger amount of explosive mixture will be formed back in the tube or line, as the distance the explosive mixture will be forced back depends upon how quickly the equalizing element, the pressure developed by the compression of the gas in the tube, overcomes the effect of the back pressure.

124 In connection with this phenomenon of the creation of explosive mixtures in tubes and lines due to back pressure, it seems desirable at this time to call attention to the decidedly dangerous influence leaks in the supply lines can have in cases where a dangerous mixture is being forced backward in one tube.

125 With the knowledge that the back pressure set up by the obstruction of gas flow from a tip can return a dangerous explosive mixture into the tubes and supply lines, and further with the realization, now apparent at least, that none of the unequal pressure blowpipes is free from the possibility of the development of a flashback, it can be readily seen that no blowpipe on the market today that operates under unequal or unbalanced pressures can with certainty be declared absolutely safe. This is especially true, as it has not as yet been established how small an opening will have to be before it will cease to propagate a flame through an explosive mixture of oxygen and acetylene. It is known, though, from the investigation carried on to determine the feasibility of employing acetylene lamps for mine use, that the opening is very small. That the return of such explosive mixtures can be caused is evidenced by reference to Table 3.

126 Recognizing the dangers coincident with blowpipes operated under dissimilar pressures, attempts have been made to minimize the return of dangerous gas mixtures by incorporating check valves in the blowpipe construction. While such desires are commendable, they are in their method of fulfillment of doubtful value in that there is no assurance that the valves will always be operative. Failure or loss of life in springs, or pieces of dirt or scale, or corrosion of a stem or seat may readily cause a check valve to fail to function. Its utility, then, is very questionable.

127 Another group of manufacturers has sought safety in action by attempting a design based on delivering the gases under equal pressures. Possibly also they have realized to some extent the

effects of excess oxygen or acetylene on the development of flashback phenomena. A very careful study of their letters patent shows, however, that they have not completely understood such phenomena, their cause and effects and the methods necessary for their suppression. It is therefore not surprising that not one of the so-called equal-pressure blowpipes proves actually to be such. The data of Table 3 show this beyond a doubt. They are therefore at times subject to the same flashback difficulties and the same lack of safety in operation, their differences being only a question of relative degree and subject to the principles that have already been stated.

128 In mentioning above the things that seemed to be responsible for the flashback, specific reference was made to the overheating of the tip and head. All blowpipes can be made to flash back from such a cause. How quickly they will do so seems to depend entirely on their design. It is certain that water-cooled heads can assist in restraining such flashbacks. It is also apparent that a tip and head made of metal of high thermal conductivity, pure copper for example, are beneficial. In fact, any construction that tends to maintain a cool gas flow seems to be helpful, and it is probable that in that fact lies the explanation for flashbacks caused by overheating of blowpipe parts.

129 It will be recalled that Davy's experiments showed that the critical-mixture proportions were affected by changes in temperature, that a slight increase in temperature made what is normally a non-inflammable mixture inflammable. It is known also that the velocity of propagation of flame is increased with increases in temperature. It is evident, then, that under the influence of a high temperature there is an increasing tendency for even a blowpipe designed for operation with excess acetylene to flash back on the development of back pressure.

130 The real cause for flashbacks developed under the influence of high temperature, where back pressure does not necessarily exist, is probably due to the sudden heating to the ignition point of a large volume of gaseous mixture. The tip and head become extremely hot for their entire length and subject a relatively large volume of gas mixture to the effects of the high temperature. The result is naturally a detonation. The suddenness and violence with which such flashbacks occur and their destructive effects on the tip ends as shown in Figs. 27 and 28 are evidence that the burning gaseous mixture has been subjected in reality to a detonation.

131 The difference in action that takes place in the two forms

of flashback, namely, those caused by back pressures and those caused by detonation of the gaseous mixture, can probably be best illustrated by calling to mind the familiar statements concerning gun cotton, a material that, lying loosely and open to the air will burn harmlessly if ignited by flame, becomes a most violent explosive if fired by a percussive detonator. It is a question of the rapidity with which a large volume of the mixture is ignited.

132 If then the sudden firing of a relatively large volume of a gaseous mixture produces a detonation, it is quite evident that the rate of flame propagation is exceedingly high. For an oxygen-hydrogen mixture it has been reported as approximately 9315 ft. per sec.; for a carbonic oxide mixture, a relatively slow and inert combination, at about 3575 ft. per sec.[1] Figures for acetylene do not seem to be available, but it is certain that they will be nearly as high as those for hydrogen.

133 The futility of attempting to prevent flashbacks from this cause by ejecting the mixture at a high velocity is very apparent. Knowledge, however, of the cause of the action indicates that anything that tends to keep the gaseous mixture cool tends to delay the development of such flashbacks.

134 Having discovered the causes of the development of the so-called flashback phenomena and showing that the safety of a blowpipe is dependent upon the causes back of these same phenomena, it is next in order to inquire as to what possible effect the cause of flashbacks may have upon the gas ratio, that is, upon the economy of operation and upon the process of autogenous welding.

135 The questions are hardly proposed before their answers are apparent. It cannot be expected that a blowpipe whose gaseous-volume mixture is constantly changing from the one-to-one volume ratio to an excess carbon (acetylene) content or an excess oxygen content can maintain a constant gas ratio or a truly neutral flame. If this is so, it is hopeless to expect such blowpipes to be capable of producing unoxidized or uncarbonized welds.

136 The controversy on the causes for unsound welds has of course been two-sided. It has been said that even experienced welders excuse the shortcomings of their work by claiming that the blowpipe or the welding rod was not good. But generally up to this time the condemnation has been placed principally upon the welder, and there is no doubt in certain cases rightly so.

137 In certain quarters the oxy-acetylene weld has been severely

[1] D. Clerk, The Gas, Petrol and Oil Engine, p. 116.

criticised. As a result of this criticism it has been suggested that all welders should receive special training, that the quality of their work be constantly checked by tests, and that they be licensed. Extensive metallurgical studies of welds and attempts to find non-oxidizing welding rods have been made. All these are good and useful expedients, but if in the first place the operator is not furnished with a blowpipe that has been designed with a correct interpretation and application of fundamental physical laws, it cannot be expected that sound welds will be forthcoming, no matter what other expedients are employed in order to increase the efficiency of the weld.

138 The oscillatory motion given to the blowpipe in the pro-

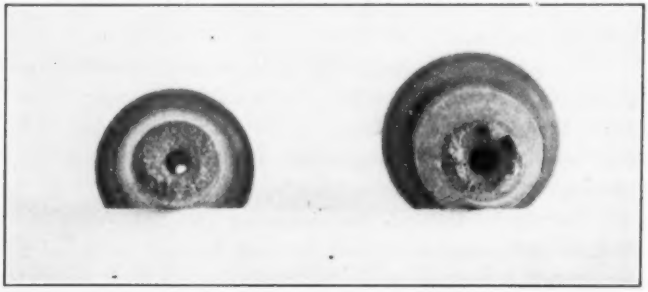


FIG. 27 EFFECTS ON TIPS CAUSED BY FLASHBACK

cess of welding and the trembling of the operator's hand, by causing a constantly varying back pressure as the blowpipe tip approaches or recedes from the surface of the weld, insure a constantly fluctuating gaseous mixture with a predominating tendency as indicated above toward an oxidizing flame.

139 In the more careful welding operations very great pains are usually taken to burn out oxide or slag pockets, etc., with the result that the tip end is forced down into a confined space. Even with such care welds are found to be oxidized. The bottom of the V of a weld is almost always oxidized, while the upper portion may show more or less of the clean cast metal. It has been known for years that a shaky-handed welder could not make sound welds. It has also been known that certain blowpipes, as regards flashback, were particularly susceptible to the nearness with which the blowpipe tip approached the welding surface. Further, some at least have

known that after placing an ignited and properly adjusted blowpipe flat on a surface and then gradually raising the head, keeping the tip upon the surface meanwhile, the blowpipe would flash back after the head had been elevated to a certain angle. These are some of the facts that indicated that there was being developed at such times the detrimental back pressures.

140 All the data available as a result of the investigation bear out the logical development of the above theory: the high gas ratios during welding, the difference in gas ratios when identical tests are performed by different operators, the lower gas ratios when

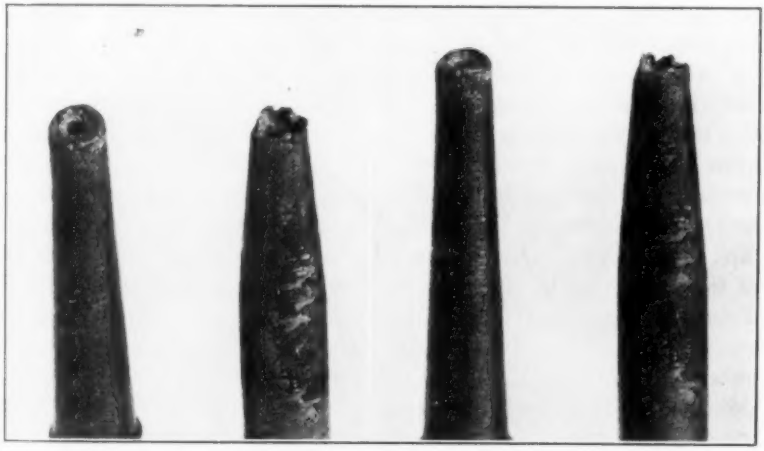


FIG. 28 EFFECTS ON TIPS CAUSED BY FLASHBACK

a blowpipe burns freely in air (see Table 5), the fluctuations in volume flows as determined by flowmeter readings taken every few minutes with gas pressures maintained within a few hundredths of a pound per square inch throughout the entire test period. Most convincing of all is the entirely unsatisfactory character of the entire group of test welds made during the investigation, a group of welds that probably was made with greater care than has ever been bestowed upon any other such set.

141 All past experience, including the experience gained during these tests, points to but one convincing conclusion, namely, that a blowpipe designed to be absolutely free from flashback caused by any form of obstruction, under all working conditions, will also be

the eminently safe blowpipe and the blowpipe that will with ordinary care produce sound welds. Such a blowpipe will be one so designed that, under all conditions of operation even to complete blocking of the gas exit at the tip end, there will be maintained a one-to-one volume delivery of each gas, at identical pressures. In comparison with these fundamentals there is no oxy-acetylene blowpipe on the market today that fulfills the conditions.

142 *High Gas Ratio in Welding.* A few other points are of interest in connection with the welding-blowpipe investigation. It was mentioned above that one of the most noticeable outstanding features which was the first to be investigated in the attempt to solve the apparent discrepancies in welding-blowpipe data, was the high gas ratio obtained during welding operations.

143 Attention has also been called to the fact that such ratios were lower when the blowpipe burned freely in air. The reason for this is of course now apparent. But it will be of interest to note that even such values were relatively high, and this notwithstanding the further fact that in a majority of gas-ratio tests the neutral flame was set in the presence of and met the approval of experienced blowpipe operators, representatives of the manufacturers. Consideration of these facts led to the belief that what is called the neutral flame is more or less masked by the operator's personal equation, that is, what he considers a neutral flame to be. Further, continued experience indicated that when the gases were issuing at a high velocity, the extreme ragged edge of the inner cone seemed to interfere with the determination of the neutral flame. It became apparent, then, that the general tendency was toward setting an oxidizing flame. This was strengthened by the knowledge that all treatises and books of instruction that were available describe the neutral flame as having been secured when the intensely white flame or inner cone is "sharp in outline . . . symmetrical and smooth." (Fig. 29.) The trouble with this statement lies in the emphasis given to the words "sharp" and "smooth." In adjusting the burning gases to produce a neutral flame it seems to have become an ingrained action upon the part of the operator to cut the acetylene-gas flow back until the entire inner cone is "sharp" and "smooth," with the result, as exhibited by the gas-ratio tests of this investigation, that the flame has actually been made quite oxidizing.

144 Having come to this conclusion, the idea of examining the flame spectroscopically presented itself, and for this purpose, because of its convenience, a replica diffraction grating was used. The use of

this method suggested itself as it was recognized that in a carbonizing flame there must exist unburned, incandescent solid carbon particles which would superimpose their blurred continuous spectra over the line spectra of the flame. Such proved to be the case, and it was found, as anticipated, that probably too much attention has been paid in the past to setting the inner cone to too sharp a contour, especially on the extreme end. The results of a series of gas-ratio tests run with the use of the gratings are shown also in Table 5, and it will be noticed that they show very marked changes in values.

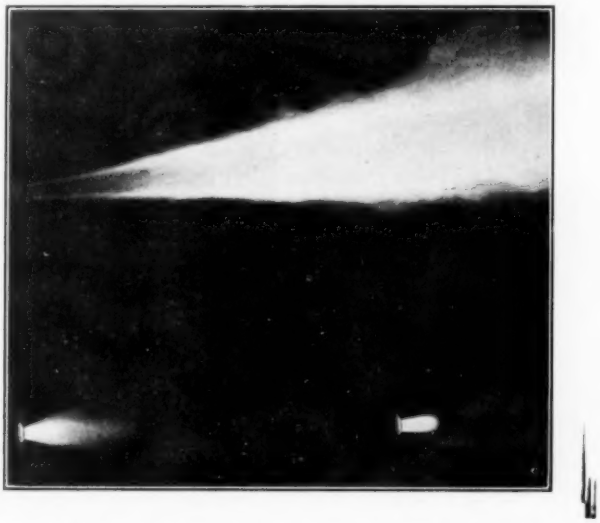


FIG. 29 EFFECT OF PERSONAL EQUATION IN DETERMINATION OF NEUTRAL FLAME

145 This table shows that practically any of the present blowpipes can be made to produce a neutral flame and burn equal volumes of oxygen and acetylene if the flame can burn undisturbed in the air. But, as stated above, none of them can maintain such a flame during the welding process.

146 *Strength of Welds.* Referring again to the strengths of the welds executed during these tests (see Table 6), it will be seen that the second plate welded during each test invariably showed higher strengths. It will be remembered that to counteract the effects of expansion the 2-ft. test welds were made by welding two 1-ft.

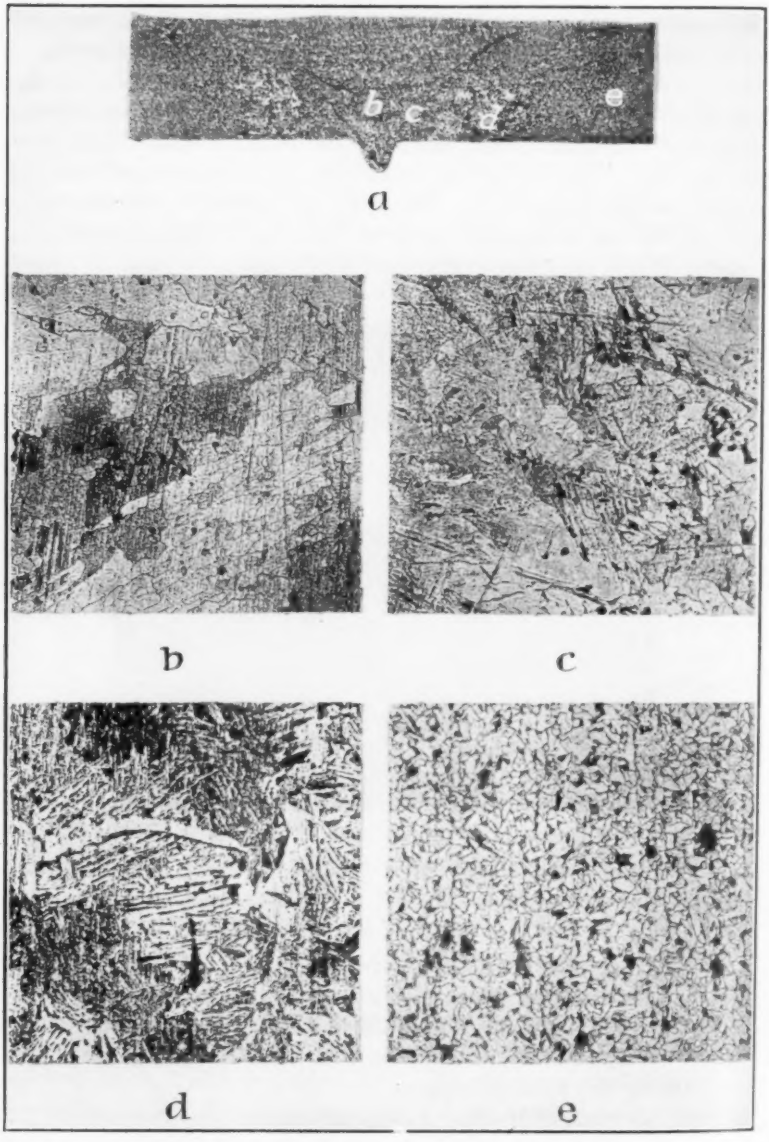


FIG. 30 PHOTOMICROGRAPHS OF TYPICAL WELD

Etched with 2 per cent alcoholic acid. × 50.
a Location of photomicrographs
b Fused-in metal of weld
c Junction of weld and plate
d Overheated metal of plate
e Initial or unchanged condition of plate

lengths of plate. These plates were supported on a heavy iron casting that contained a channel throughout its length parallel to and directly under the line of welds. This channel caused the flame of the blowpipe to return under the plates being welded and thus preheated to some extent the second plate. Further, it caused a decided heating of the near end of the base casting. The explanation for the higher strength shown by the second plates probably lies in the more uniform preheating of the second plate and the greater annealing effect produced by the heated base casting, the latter causing a release of the tensile strains resulting from the contraction of the metal along the line of the weld.

147 The average strengths and average included angle of bend are given in the above-mentioned table (Table 6) for what they are worth. It is very probable in the light of present knowledge of the requirements of blowpipe design that some new ideas will be forthcoming concerning the average strength of oxy-acetylene welds. Finally in Fig. 30 are exhibited photomicrographs showing the effect upon the grain structure of the material, in this case mild steel, of the autogenous welding process, effects not necessarily detrimental when properly performed, but, as exhibited by the photomicrographs, instructive.

DISCUSSION

S. W. Miller. The paper is the result of probably the most complete set of tests on oxy-acetylene welding torches that has ever been made. There are obvious reasons for not publishing many of the data that were secured during the tests, and some of the conclusions may be based on data that are not published. These conditions make it difficult to speak with confidence of the conclusions drawn.

It appears quite evident from the published data that the variations in the cutting test were much more serious than anyone had ever expected. It must be remembered, however, that while it is necessary, as well as desirable, to have economical apparatus, commercial considerations must also be considered, so that, for instance, while it would be advantageous to have a proper size of tip for each thickness of metal, it would hardly be possible to use them in practice. Much of the cutting that is actually done is on rusty material, or that which is coated with scale of various kinds, and there is often a serious question as to the advisability of removing all of this foreign material. This the author has referred to but as he also says,

it would require more tests directed along the lines indicated by the present ones to determine just what should be done to develop the most economical and satisfactory torch. It is entirely probable that a compromise will have to be made in order to secure the most satisfactory results from all points of view. The present tests furnish a good basis on which to work.

It is with the welding torch, however, that the writer desires to deal particularly.

Any tensile test on a non-uniform or non-homogeneous piece of metal must be interpreted with caution. It is too frequently assumed that a weld in steel plate is quite of the same character as the plate itself. This is very far from the truth, as those who have investigated the matter well know. Assuming that the weld is properly made, and thorough fusion obtained between the plate and the weld material, it must be remembered that the weld is a casting, that it is of necessity more or less over oxidized, and that metal in this condition, even if of the same mechanical composition as the plate, has very different physical qualities.

Steel plate varies in quality very much. It may run in tensile strength from 48,000 to 68,000 pounds. Similarly the carbon content may be as low as 0.08 per cent, while in ship plate it may, in the thicker sections, be as high as 0.3 per cent, which is necessary in order to get the reqired tensile strength. The other usual elements, manganese, sulphur, phosphorus and silicon, may vary within quite wide commercial limits.

The effect of these elements, taken singly, is not known except in a very general way. It is known in a general way that high-carbon steel is more difficult to weld than low-carbon steel, but the quantitative effect of carbon is still unknown.

The same is true of the other elements commercially present in steel. When it comes to combinations of the various elements, nothing is known, for instance, as to the effect of varying manganese in the presence of a constant amount of carbon. It is quite probable that other elements which are not usually analyzed for, such as nickel, chromium and copper, and which are becoming more frequent in commercial steel, even when their presence is not desired, have some effect, but it is not known what this effect is.

One other element, oxygen, is almost never tested for, and yet the writer believes that its presence is by far the most detrimental of any element, both in the steel and in the weld. From the method of making a weld, it would seem impossible to produce a metal

that would not be over oxidized, even omitting entirely the possibility of excess oxygen in the welding flame. The hydrogen flame itself causes oxidation, because the H_2O vapor formed dissociates at the temperature of the melted steel, and the oxygen combines with the latter.

It will, therefore, be seen that it is not possible to make a weld of as good quality as that of properly made steel, and that precautions must be used to maintain the conditions during welding reasonably constant, and that the work must be done by those who have been properly trained, if satisfactory results are to be obtained. It is not, however, to be understood that satisfactory results for most purposes cannot be obtained, if proper precautions are followed. As a matter of fact, experience has shown that when these precautions are used, no trouble is experienced.

The quality of the welding material and the effect on it of different chemical composition have never been investigated as far as the writer knows, except in the most general way. It has been found that, generally, satisfactory welding wire contains only a small amount of carbon, say not over 0.10 per cent. On the other hand, there are some objections to the use of this material; the principal one of which, in the writer's opinion, is that in the weld, it is red short, and therefore unsuitable for any place where the weld is liable to be overheated.

The writer has found that three and one-half per cent low-carbon nickel steel of the standard S.A.E. analysis is not red short, but can be forged at any temperature. It also gives a higher tensile strength in the weld than ordinary material, although its elongation is somewhat less.

These matters are spoken of because it is clear that, with a given welding material and other things being equal, variations in the tensile strength of the plate will give variations in the results of tensile tests of welded pieces, just as will variations in the welding wire. It has been clearly shown that, in the case of 50,000-lb. tensile strength plate and a weld made with nickel steel, of the same section as the plate, the plate itself will break under tensile test, while with 60,000-lb. tensile strength plate, the break will occur in the weld. Therefore, it is necessary to know the physical qualities of the plate, before drawing any conclusions as to the efficiency of the weld.

It is possible, of course, by reinforcing a double-V weld sufficiently, to make it stronger than the plate, but this is not possible

in the single-V weld. In the latter case, the heavy reinforcement made by adding metal on the top of the weld tends to weaken it rather than to strengthen it, due to the fact that the load is eccentric. There is, still further, a practical condition which affects the results even when the weld is apparently sound, as in many cases there are defects, at times of microscopic size, at the bottom of the V. These defects are in the form of cracks, which tend to propagate under stress, and cause rupture at a lower load, than if they were absent. In the case of double-V welds, these defects, if they exist, are in the neutral axis of the piece, and therefore have less effect than in single-V welds.

There is one more thing which has a greater bearing than any of those mentioned on the strength of weld, and that is whether the weld is sound or not. Evidently, if thorough fusion has not been obtained along the sides of the V, no conclusions can be drawn as to the tensile strength or other physical properties, and the writer believes that in every test the condition of the weld with regard to this point should be recorded.

It is evident that in the case of a single-V weld, if the bottom of the V is not thoroughly fused, the condition of eccentric loading is aggravated.

In the paper, there is no information given as to the quality of the welds. The writer would suggest, therefore, that great caution be used in drawing any conclusions from the tensile tests, as to the efficiency of the torches tested.

In order to show the irregularity more clearly than the tables do, the writer has plotted the results of the gas ratios against the tensile tests, as given in Tables 5 and 6, using the test series 1(a), 1(b), and 2. (See Fig. 31)

It is rather curious to note that the highest oxygen to acetylene ratio, 1.41, in test No. 2, obtained with torch No. 4, gave the highest tensile strength, 53,500 pounds. The next highest tensile strength, 52,100 lb., was obtained with torch No. 13, with a gas ration of 1.02.

Another instance is where, with the same gas ratio, 1.26, a tensile strength of 32,100 was obtained with torch No. 3, while with torch No. 11, in three-quarter-inch plate, a tensile strength was obtained of 45,100 pounds. These are not isolated instances, as an examination of the plotted results will show. Further there is a greater difference between the results in individual tests than would appear to be reasonable. For instance, with torch No. 3, test 1(b), the tensile strengths are given as 27,900 and 37,900 lb., a difference of over 28 per cent based on the smaller figure.

With torch No. 11, test No. 1(b), there is also a difference of over 31 per cent. In test No. 13, 1(a), there is a difference of over 32 per cent. Certainly these differences cannot be ascribed to differences in torch peformance. At least nothing is given in the data that would indicate this, and from the writer's experience, he would certainly question any such interpretation.

It might also be noted that the variation in the bending angle is equally erratic with the tensile strength.

With regard to the general averages, there is not enough differ-

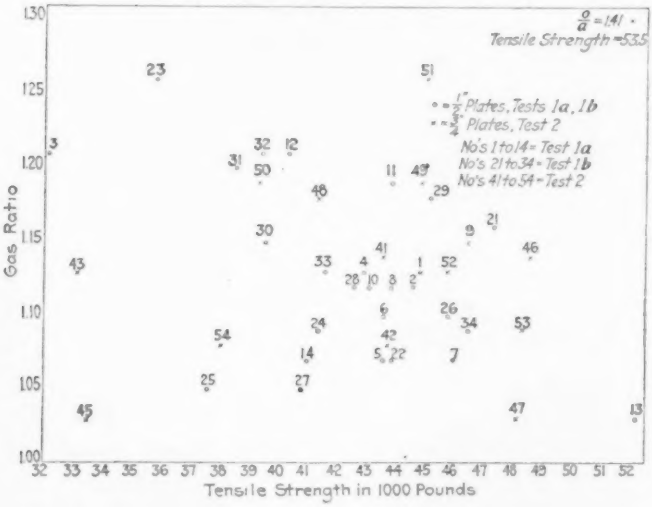


FIG. 31 VARIATION OF GAS RATIO WITH TENSILE STRENGTH

ence between them in tensile strength to warrant any conclusions as to differences in operators or differences due to differences in pressures used in the torches, except that the operator that made tests No. 1(b) did not obtain quite as high results as the one that made test No. 1(a). It is, however, to be noted that in many of the individual tests with some of the torches the former operator got better results than the latter. For instance, with torches No. 1 and 6 and 14.

The selected figures appear to justify the conclusion mentioned above, that there is nothing in the tensile or bending tests that should be used as a basis for distinguishing between the relative values of the torches tested.

With regard to the gas ratio, the author states in Pars. 140 and

143, that the torch ratio, that is, the ratio of oxygen to acetylene, (which is given in the tables in the form of unity and a decimal, the actual ratio being, for instance, 1.08 to 1.0, the figure 1.08 being given in the table), is higher than it should be.

From the writer's experience, this is true. In all the tests he

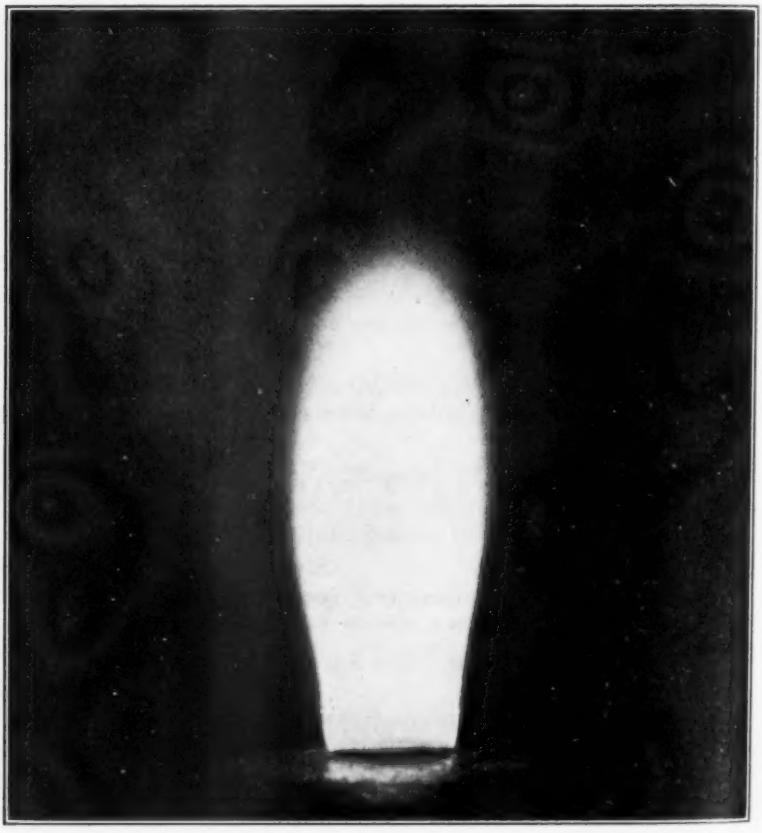


FIG. 32 NEUTRAL TORCH FLAME SYMMETRICAL WITH AXIS OF TIP MAGNIFIED 5½ TIMES

has made, he has never met with any such ratios as with torch No. 4 in tests Nos. 2 and 5(b), 1.41 and 1.29, and with torch No. 8 in tests Nos. 5(a1) and 5(b), 1.27 and 1.43.

There are others that are in the neighborhood of 1.26, and the writer is inclined to believe that the operator was careless in adjust-

ing the flame. There is, however, one other possibility. In many torches the small white cone, which is the welding flame, is not symmetrical with respect to the axis of the tip. There may be several causes for this, which it is not necessary to study now, but in such tips it will be found that as the proportion of acetylene is decreased

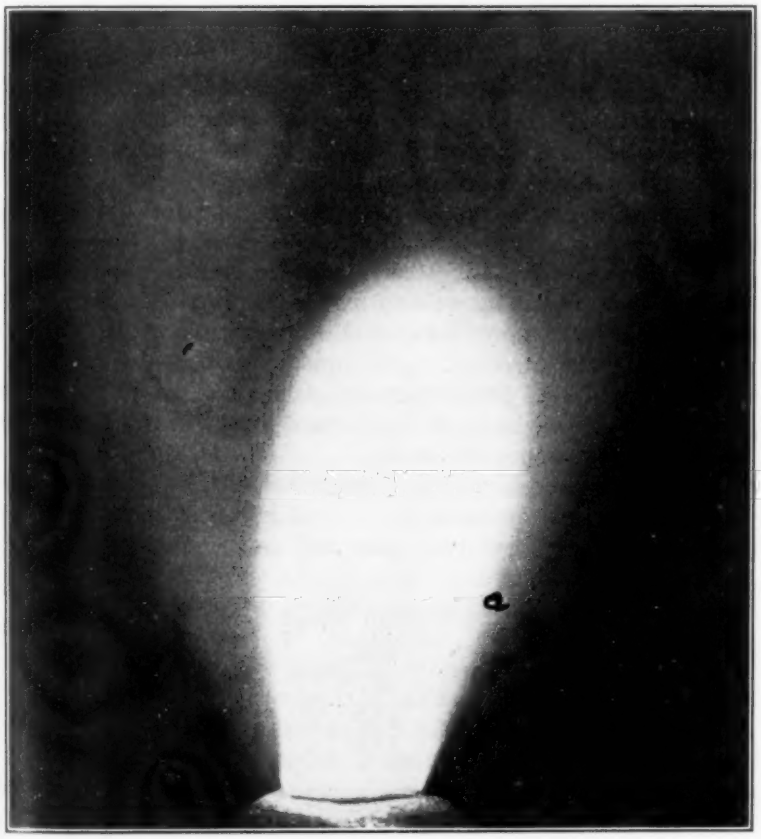


FIG. 33 TORCH FLAME UNSYMMETRICAL WITH AXIS OF TIP MAGNIFIED 5½ TIMES; *a* SHOWS FEATHER OF ACETYLENE

further by increasing the oxygen or decreasing the acetylene, a feather of acetylene will persist on one side of the flame, so that when one side of the welding flame is neutral, the other side will be excess acetylene, as shown in Fig. 33.

Evidently, if enough oxygen is turned on to neutralize the

feather on one side, there will be an excess of oxygen on the other. Even this does not explain the discrepancy between the ratios, as obtained by weight, and obtained by the flow meters, as for instance, with torch No. 4 in tests Nos. 2 and 5(b), 1.41 and 1.16, 1.29 and 1.10.

In his book on oxy-acetylene welding, the writer stated with regard to the neutral flame as follows: "When the oxygen pressure is turned on sufficiently to just destroy the double cone produced by deficiency of oxygen, the flame is neutral." This was written in 1915. Since that time he has found that in order to get as nearly a neutral flame as possible, it must be observed through dark glasses, as the intense white light affects the eyes to such an extent that the judgment of the operator may not be correct.

Another matter that at times produces incorrect results, not only in adjusting the flame, but in the entire welding process, is the sight of the operator. Altogether too little attention has been paid to this point. Successful welding requires acute vision, and unless this exists in the operator, the minute troubles which cause defective welds will not be observed.

Frequently, also, glasses that are too dark are used. The criteria for these conditions have not yet been defined, but they must be taken care of in important work.

In a test made on four welders with the same torch and tip, the acetylene pressure being kept constant within the limits of a special test gage, the oxygen pressures, as recorded by a similar gage, were as follows:

	Without glasses	With glasses
Welder No. 1	13.5	13.3
2	13.7	13.5
3	13.5	13.2
4	13.4	13.6

These tests were made with commercial regulators which are never absolutely accurate, they being the weakest point of the whole oxy-acetylene system, as they do not maintain a constant reduced pressure with a varying initial pressure.

There is no doubt, however, that the author's statement is correct that the tendency is to over-oxidize the flame. A trifling excess of acetylene is not nearly as dangerous or objectionable as a similar excess of oxygen, because if there is a carbonizing effect of the excess acetylene, it is neutralized by the burning out of the carbon by the heat, while the excess oxygen is not so disposed of.

Another point which has not been mentioned as far as the writer has noticed is that commercial oxygen is not pure, and no allowance has apparently been made for this in Table 5. This, of course, would reduce the proportion of oxygen, and therefore the figures given in the table.

It is equally true that acetylene taken from tanks contains some acetone vapor, and that this has also not been allowed for. But acetone does produce some heat, while in the case of liquid-air oxygen, the chief impurity, nitrogen, does not.

The author also states in Pars. 140 and 143 that the gas ratios are higher during welding than when burning freely in the air. It does not seem that the figures bear out this statement. It is true in individual cases, but it is untrue in others. As far as averages are concerned, they are so close that such a conclusion would seem to be unjustified. Averages of columns 2 and 3, Table 5, are 1.12 and 1.13. Of column 6 and column 5 of the same table they are 1.116 and 1.12, also when columns 5(a1) and 5(b) are compared with 1(a) it will be found that in column 5(a1) there are 7 ratios larger than those for the same torches in column 1(a) and 7 smaller. In column 5(b) there are 5 that are larger and 9 that are smaller. Surely there is not enough difference on which to base any conclusions.

Further, in the tests with gratings, the highest ratio of any in the table was obtained with torch No. 9 in test 5(a1), although it is true that in all the other cases except one torch, No. 11, the grating test showed lower ratios than the tests without them.

There is another point that should be considered carefully, and that is the thoroughness of the mixture. If the tip construction is such that the gases are stratified or otherwise improperly or not thoroughly mixed, the condition of the flame will be bad for welding, even though the gas ratio is theoretically correct. When it is considered that the time of passage from the mixing chamber to the end of the tip is usually considerably less than 0.001 of a second, it is almost unfair to expect a perfect mixture to be obtained. This time can be readily checked by the fact that in a welding torch the exit velocity of the gases is in the neighborhood of 400 ft. per sec., and in a tip 4 in. long, it would only require one-twelve-hundredth of a second to pass from the mixing chamber to the exit. This time is shorter, of course, in shorter tips, but as the bore of the shorter tips is smaller, the mixture is probably as good in one case as in the other.

Considering all the foregoing, it is entirely possible to have free

oxygen and acetylene in the flame, even when the gas ratio is theoretically correct.

In Par. 139, the author states that the bottom of the weld is almost always oxidized while the upper portion may show more or less of the clean cast metal. It has not been the writer's experience that the bottom of the V is any more oxidized than the upper part. As far as ferrous oxide is concerned, his observation is that the bottom of the weld is cleaner when the weld is sound. This ferrous oxide appears in the shape of small round dots which of themselves do not have any particular effect on the strength of the weld. The rupture in welds is always either at defects which are clearly visible, such as films of oxide, slag, etc., or at the grain boundaries, even where there are no visible defects.

It is probable that the cause of these intergranular ruptures is not ferrous oxide, but ferric or magnetic oxide, and there would seem to be no reason to doubt that the presence of large amounts of ferrous oxide is indicative of the presence of other oxides, possibly not always of iron, or of other impurities which occur at the grain boundaries.

It would be fair to assume that if a torch in which a higher oxygen gage pressure is used will give an excess oxygen flame during welding conditions, one which uses an excess acetylene pressure would, under welding conditions, give an excess of acetylene, as it is well known that if the tip of a torch of the latter type be partially obstructed, the flame becomes excess acetylene even to such an extent as to be smoky when the obstruction almost entirely shuts off the opening.

In Par. 146 the author states that the second plate welded during each test invariably showed higher strength. This statement is not true of torch No. 4, test 1b; torch No. 6, test 1b; torch No. 11, test 1a and 1b; torch No. 12, test 1a and of torch No 11, test 1c.

In the cases in which the statement is true, the writer doubts very much if the explanation given is sufficient. He believes that the fact is that the differences in tensile strength and bending angle are simply fortuitous, and due to differences in the quality of the welds. The great trouble with all welds, and the principal reason for their irregularity, is that the welder does not thoroughly fuse the metal. This is usually due, on his part, to a lack of appreciation of the necessity for such welding, or to a lack of instruction as to just how to proceed. It is rare to find a welding shop in which the welders are regularly tested as to their ability to do good work in plate steel, by having them make test pieces at regular intervals.

It is not necessary to have testing machines to determine this. Pieces can be clamped in a heavy vise, or on an angle, and broken with a sledge. The amount of bending before cracking is an index of the ductility of the weld, and the appearance of the fracture is a good criterion of the soundness of the weld. The writer feels confident that, if this test were introduced in all welding shops, a greater improvement in the quality of the work would be obtained, than by any other method, and it would be obtained in a very short time.

STUART PLUMLEY.[1] Had the author confined himself to the facts and figures pertaining to his test, it would without doubt have been of much value to the industry, but unfortunately, apparently based upon results produced during the test, the author has reached a number of conclusions which are not justified by the facts involved and which evidently show a lack of knowledge of present practice in the industry. The most prominent of these conclusions may be stated in a few words as follows:

First, that the maximum thickness of metal which may be cut with an oxy-acetylene torch of standard design, is about 12 inches.

It is surprising that such a conclusion should be reached in view of the fact that it is common practice with certain corporations who carry on heavy cutting, to cut with the oxy-acetylene torch heavier sections than 12 inches. These thicknesses of steel vary all the way from 12 up to 30 inches. The author could not have been familiar with these operations and must have reached the conclusions which he did, based entirely upon the results of the test.

Cutting is divided into two general classes — hand cutting and machine cutting. Hand cutting consists of the miscellaneous work which is ordinarily done with a hand cutting torch and in most instances metal heavier than 6 or 8 inches is not attempted with a hand torch except in the case of large steel risers. In such instances it is usual not to attempt to cut the riser clear through but to break a part of it after it has been partly cut.

With a hand torch the average manufacturer does not usually supply tips having a large orifice for cutting, because so little heavy cutting is ordinarily accomplished with these torches. Some manufacturers do, however, supply these tips whenever such cutting is to be done with a hand torch, and in such instances the larger tips provide much larger orifices for the cutting oxygen. These tips are not special in any manner, but are standard tips for use with hand cutting torches. As an example, the largest tip which was used in

[1] Davis Bournonville Co., Chicago, Ill.

the test upon one type of torch with which the writer is familiar was known as a No. 5 The area of the oxygen orifice of this tip is equal to 0.0075 sq. in. The largest size of standard tip, known as No. 10, for use with this torch, has an oxygen orifice equivalent to 0.0254 sq. in. The largest tip therefore has an area of oxygen orifice which is over three times the area of the largest tip of this manufacture used during the test.

The author does not state that his conclusions are based upon an intermediate size of tip, nor does he seem to be informed that such tips are in daily use. The performance upon which his conclusions are based is not the maximum performance of the torch which he used and the conclusion is therefore incorrect.

The depth which can be successfully cut with the oxy-acetylene torch is dependent largely upon the volume of oxygen projected against the metal, and this volume of course under a fixed pressure is dependent upon the area of the ports from which the gas issues. It has been determined that with a fixed area of port, increasing the pressure of the oxygen is not sufficient, but for heavy cutting the area of the port, and hence the volume of the oxygen, must be increased proportionately.

The writer has seen a similar torch with a No. 8 tip produce a cut in a 54-in. steel billet, the maximum depth of which was 36 in. of flowing slag. No attempt was made to cut this billet completely through. Several cuts were necessary to sever the riser from the billet, but at one period during this cutting the torch made a cut which was 36 in. in depth and was sustained through a length of several inches. Such cutting, however, is as a rule suited more to a machine than to a hand torch. Machine torches are upon the market having tips for various purposes and of many designs. For heavy cutting a large tip is used. The diameter of the ports and the size of the preheating flames are proportioned to the work which is to be accomplished. Apparently none of these machine cutting torches or tips were used by the author in this test.

On the other hand, machine torches are also used for cutting lighter work such as dies, cams and other irregular shapes. The thickness of these may vary from 1 in. up to 4 or 5 or perhaps 6 in. and in each instance suitable tips are used for the work in view.

The author criticizes the use of too large preheating flames. Apparently he was not aware that common practice in machine cutting torches includes the use of much smaller preheating flames than with hand cutting torches. His test propelled hand cutting

torches by mechanical means, but instead of using machine cutting torches developed for the purpose, he applied hand cutting torches to a use for which they were not intended. It is obvious that with a hand cutting torch, where a large proportion of the cutting done is upon old, rusty, scale-coated metal, the preheating flames must be larger than is necessary in machine cutting torches where in most instances new metal is used and the torch is held in a fixed position with relation to the metal.

Second, a good deal of the discussion pertains to the alleged incorrect design of apparatus and to oxygen and acetylene pressures delivered to the torches. Reference is made to torches having a high oxygen pressure, others having a high acetylene pressure and those operating under equal pressures.

The author has apparently based all of his remarks and conclusions upon pressures obtained within the hose supplying the torches. In other words, his remarks apply to what is generally known as the regulator pressure and not to the actual pressures existing within the torches or tips where the gas enters the mixing ports.

Pressures which really determine the type of torch are not the regulator pressures at all but are those existing within the ports as they approach the mixing chamber and these pressures are dependent upon the area of these ports as well as the pressures at which the gases are delivered by the regulators. In other words, a torch having a higher oxygen than acetylene pressure shown upon the regulator gages might in reality be an equal-pressure torch if the areas of the ports were so porportioned. These pressures within the torch or tips adjacent the mixing chamber do not seem to have been given sufficient consideration by the author.

Third, a large part of the paper is given over to a discussion of flashbacks and far more emphasis is placed upon flashback prevention than is really warranted by the existing facts. It is true that a flashback is very annoying to the operator when in the midst of particular work. It is also true that such a flashback may produce a hazard under certain conditions. Normally, however, with a torch in proper operation, flashbacks do not occur and with most torches the hazard is extremely slight. If a flashback does occur during welding, it is usually because of a lack of acetylene, an overheated torch or an accumulation of dirt or soot within the gas ports. It may also be produced by obstructing or plunging the tip of the torch in the molten metal. Even such severe treatment will not always

produce a flashback and in fact many tips may be indiscriminately plunged into the metal and still not result in a backfire. It is obvious, however, that such treatment is really abuse of a torch.

Perfectly balanced pressures are not attainable in any ordinary field apparaturs. Both oxygen and acetylene regulators are essentially reducing valves, and reduce higher pressures within cylinders or generators, to lower working pressures at the torch. While these working pressures are fairly even and constant, they are not absolute and cannot be under practical working conditions. It is therefore out of the question to concieve of a torch in which the gas pressures are exact. It is also obvious that the restriction of the flow of the mixed gases through the main port of the tip is also a variable and under working conditions will always be governed by the proximity of the torch to the metal, the action of a bit of slag against the opening of the port, and to the confined space in which the tip is being operated. These conditions exist in connection with all work and are constantly being changed by the operator as he draws the torch away from the molten metal.

To produce conditions, or to permit them to exist, which cause a backfire, is due largely to lack of skill on the part of the operator and is an abuse of the torch. The possibility of making a torch backfire and the fact that when wrongly used it may backfire does not seem to be proper justification for the author's statement that such a torch is not safe or that it will not produce a sound weld.

Fourth, the author gives values for the strength of welded specimens made during the tests which are much lower than is usual in the common paretice of the art in this country. There are many welds being made every day having an average tensile strength which is much greater than that produced during these tests.

He also does not take into consideration many other tests of welded specimens which have been made from time to time by others interested in the process and which show greater values. Under these conditions the writer can only assume that he did not take into consideration the personal element in connection with an operator and a particular type of torch. It is a well known fact than an operator as a rule does better work with a torch to which he is thoroughly accustomed than with one with which he has had a limited experience. After all welding is an art and it must be mastered by an individual workman before he can produce good results. If he learns to weld with a particular type of torch, as a rule he can do better work with this type of torch than he can with some other

even though both types operate much in the same manner. To expect any given number of workmen to produce exceptional results with a varied assortment of torches is asking for more than seems justified. One is inclined to believe that the operators themselves found that this condition prevailed. There is also a wide variation between the results attained by individual operators. In making any series of tests such as those under discussion, it would seem that the personal skill of the operator should be given much more consideration than is apparent in this paper.

J. I. Banash. The writer cannot find himself in accord with many of the author's conclusions.

The situation of oxy-acetylene gas welding and cutting is not really quite as gloomy as might be inferred from this paper. Practical considerations must not be forgotten. Welded barrels are still used for gasoline, steel risers are still being cut off with the oxy-acetylene cutter.

A fundamental point has not been given proper consideration, although it is a most important factor in welding — the personal element.

It is claimed that expert welders and cutters from navy yards were employed. Obviously from the ultimate strength of welds produced and the limits of cuts the operators were not expert as this word is understood in the welding industry, and therefore such data can only be considered as average and not as approaching maximum efficiencies now attainable.

The average ultimate strength given in the paper is 39,200 lb. per sq. in., whereas data covering a series of recent tests on practically the same material under similar conditions, with the work done by a first-class welder, shows that with Norway iron filler an average strength of 55,600 lb. per sq. in. was reached, the lowest figure of five tests being 53,400 and the highest 57,000. This is about 16,000 lb. higher than given in the paper. With mild steel filler even better results were obtained. In five tests the ultimate strength ranged from 57,500 to 60,500 lb. per sq. in., showing an average of 59,340.

The operator does not seem to have obtained results in hand cutting for maximum thickness which are today ordinary routine in foundries, forge shops and armor-plate mills, where single cuts of 18 to 24 in. for sustained lengths, and not merely short cuts, are regular parctice as compared with the limiting maximum of 12 in. as stated in this paper.

Criticisms of engineering design of apparatus in this paper are based on deductions drawn from such tests and are not entirely justified since the capacity of the apparatus itself was evidently not fully attained by the operators.

It may be of interest to note that one severe criticism of present design should not in the writer's opinion have been made. Preheating flames on cutting blowpipes are concluded to be too large and therefore wasteful. It should be noted that hand-cutting tips were used and the devices mounted as machine cutters. It is common knowledge, derived from field experience, that hand-cutters are more efficient with larger preheating flames than are at all necessary or desirable for machine cutting.

The following comments by a well-known metallurgist coincide with the writer's opinion:

"Par. 57. The writer understands that the conclusions of the relative showing of the torches are subject to criticism, because the torches were used under conditions for which they were not adapted. A valid objection holds because of the importance of the personal equation in both cutting and welding. Manipulation means everything and, consequently, any equipment, except that which was specially designed for a given operating condition, would obviously be denied the greatest asset it could have, namely, skilled individual supervision.

"Par. 62-A. The subject matter of this paragraph is a strong indictment of the present blowpipe equipment. However, many of the irregularities, about which the author complains, can be discounted by the considerations mentioned above.

"Par. 63. What is apparently meant to be an indictment of cutting torches can be taken in the opposite manner here. It is difficult to see how a single torch could be designed to be equally efficient on all thickness of metal, granting the author's requirements for an efficient torch.

"Par. 81. Objection is raised to the last sentence of this paragraph. The statement as it stands, without further explanation, furnishes a very serious indictment of this whole test work.

"Par. 85. The sentiment of this paragraph is good only as far as it goes, which is only a short distance in the direction of stating the whole facts. The skill and experience of the welder, the quality of the welding wire, the mass and consumption of the material, and so forth, should in all fairness be mentioned in connection with a statement like that made in this paragraph.

"Par. 140. The concluding sentence of this paragraph seems to typify the dominant note in which the conclusions from this test work have been drawn. By continually referring back to the extreme nicety of the test apparatus, much more important considerations have been lost sight of. The duplication or refinement of accessory equipment is bootless without most painstaking attention to the personal element, or to the every-day considerations of practical welding."

On the subject of comparative hazards and safety in operation the writer would respectfully offer the results of a considerable experience in examining and testing blowpipes for their safety to prove that an entirely reasonable and acceptable measure of safety is afforded by present standard equipment. If this were not the case the process itself never could have reached the practically universal use which it has attained.

S. V. JAMES. The safety of torches depends on a number of items in mechanical construction and operating methods which are of more importance than flashback. The author has devoted considerable space to flashback and has made some interesting statements regarding it, but, in the writer's opinion, he has over-emphasized the hazard. In testing torches at the Underwriter's Laboratories it has been found, as the author states, that practically any torch, under certain conditions, can be made to flashback. A considerable field experience with hundreds of torches under all conditions of service fails to show any serious trouble from flashback.

H. G. KNOX,[1] discussed the methods of testing welding and cutting blowpipes at the Norfolk Navy Yard. In these tests more attention is paid to the mechanical features and serviceability of the various blowpipes, the interest being not so much in the theoretical gas consumption as in the general economies of field service. The blowpipes are examined from three viewpoints; design, performance and upkeep. Subdivisions under these headings are rated on a decimal basis, the weighted averages giving an idea of the merit of the blowpipe, although not determining its acceptance or rejection for Naval use. One or more elements of design often determine rejection.

JAMES W. OWENS,[2] in discussing also the above mentioned tests

[1] Manfacturing Engineer, Winchester Repeating Arms Co, New Haven Conn.

[2] Welding Aid, Ass't Shop Sup't., U.S. Navy Yard, Norfolk Va.

stated that all oxy-acetylene cutting blowpipes submitted for test are required to cut metal up to eight inches in thickness, this being considered the practical utility and efficient gas consumption limit. Above eight inches oxy-hydrogen is used. In addition to the tests outlined by H. G. Knox, a six month's service test is also made on both welding and cutting blowpipes.

C. J. Coberly. The method employed in this investigation for measuring the gases is capable of a high degree of accuracy, much higher in fact, than is consistent with assumptions made in regard to some of the important variables. This renders the results very deceiving, not only to the reader but even to those conducting the tests and no doubt accounts for the broad and definite conclusions drawn by the author which in the writer's opinion are unjustified by the data.

The results of the investigation in regard to cutting due to lack of control of certain variables is not of sufficient accuracy to be of great importance to the industry. Some of the important items entering into the results which were not taken into consideration are as follows:

1 *Standard of Quality.* No attempt was made to establish a standard quality of cut. The correct speed for operating the torch was left entirely to the judgment of the operator. Taking the lag of the lines left on the surface of the cut as a measure of quality, Fig. 34 shows the variation of this angle with speed and oxygen required per foot of cut as determined by tests with which the writer is familiar. An operator by judgment alone ordinarily will not reproduce a constant angle of lag closer than ± 2 deg. at a 5-deg. angle which introduces a possible error in the result of ± 10 per cent.

2 *Purity of Oxygen.* This is an item of vital importance. The oxygen used in these tests contained from 1 to 3 per cent N_2 and in addition was saturated with water vapor at the room temperature and the pressure in the flashback tanks. Fig. 35 shows the approximate effect of these impurities. Water vapor has practically the same effect as N_2, hence O_2 containing 3 per cent N_2 and 1 per cent H_2O is only 35 per cent efficient. The variation in these factors which affected the comparative results of apparatus of different makes is not known since all of the data are not available. Assuming the purity to be 97 per cent then a variation of ± 0.1 per cent in purity would vary the efficiency by ± 2.5 per cent relative

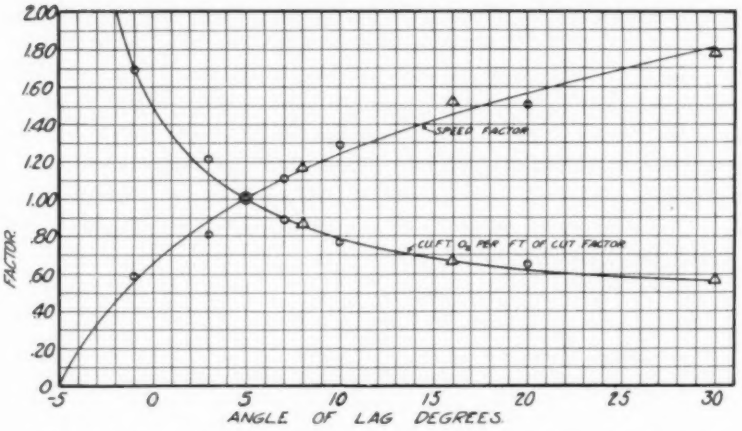


FIG. 34 RELATION OF SPEED FACTOR AND CUBIC FEET OF O_2 PER FOOT OF CUT FACTOR TO ANGLE OF LAG

NOTE: — Five deg. lag taken as standard of comparison ½-in. low-carbon steel. Oxygen orifice diameter 0.040 in. Oxygen pressure 17.22 lb. per sq. in., average. Oxygen gas consumption 38.5 cu. ft. per hr. Electrolytic oxygen and hydrogen. Angle of advance 5 deg.

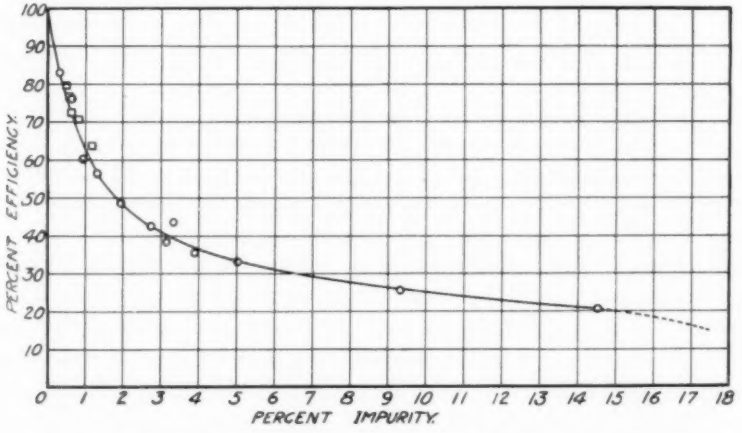


FIG. 35 CUTTING EFFICIENCY OF OXYGEN CONTAINING IMPURITIES

NOTE: — □ denotes water vapor, ○ denotes nitrogen. ½-in. low-carbon steel. Temperature of steel and oxygen 68 deg. fahr.

to the efficiency at the point. The variation of purity during these tests probably was sufficient to account for a variation of ± 10 per cent in the results.

The effect of impurities is to lower the speed of cutting and

hence, in addition to lowering the efficiency, to increase the amount of heat supplied by the flame per unit length of cut. This accounts for the fact that most of the torches tested were supplied with preheating flames that were excessive. If the same sized flame had been used with oxygen of high purity an increase of speed of over 100 per cent would be possible, which would be equivalent to using a flame of one-half the size under the conditions of these tests.

Other items of importance which affect the results to a lesser degree are given below.

3 *Temperature of oxygen.*
4 *Temperature of steel.*
5 *Carbon content of steel.*
6 *Surface condition of steel.*
7 *Barometric condition.*

These items were not investigated in detail by the Bureau of Standards and no allowance was made for variations due to these causes in computing results. No definite conclusions, therefore, are justified relative to comparative merits of the various makes of apparatus submitted except subject to the accuracy indicated above.

The limit of the 12 inches set by the author as the maximum penetration possible with oxy-acetylene is much lower than is actually found in commercial application today.

Gas ratio seems to be the determining factor upon which all of the conclusions in regard to welding torches are based. The degree of approximation of the assumed value that can be obtained even under ideal conditions is a point which has never been properly investigated and which is subject to variation from at least two causes. First, with chemically pure gases a 1:1 ratio requires the assumption that the temperature surrounding the cone of the flame is such that the dissociation of H_2 and CO_2 are complete, that the dissociation of CO is negligible and that no endothermic compounds are formed.

The work of Nernst and von Wartenburg, Mallard and Le Chatelier and others indicates that the first two assumptions are approximated but exact experimental confirmation is lacking. Considerable experimental evidence is available to show that the endothermic compounds H_2O_2, O_3, and NO_2 are formed even at temperatures materially lower than that of the oxy-acetylene flame.[1] The amounts of these compounds which it is possible to have in the

[1] Keiser and McMaster, *Am. Chem. Jour.*, v. 39 (1906), p. 96.

high temperature zone is not known but the fact that their presence can be easily detected is sufficient to show that the 1:1 ratio cannot be assumed to a high degree of accuracy. Second, the gases used in this test were not of high purity. The oxygen contained from 2 to 3 per cent N_2 and 1 to 2 per cent H_2O (saturated at room temperature and working pressure of torch); the acetylene contained from 2 to 6 per cent acetone, 1 to 2 per cent water vapor and probably other impurities not recorded.

The exact quantitative effect of these factors is unknown but obviously the formation of H_2O_2, NO_2 and O_3 will raise the gas ratio; N_2 will raise the ratio in direct proportion to the percentage present; H_2O in the $H_2O - H_2 - O_2$ equilibrium. The effect of N_2 is the only item listed above which is definitely known but this alone is sufficient to account for a variation of at least 2 or 3 per cent in the gas ratio. In view of these facts it is not logical to assume that a ratio of 1:1 is even theoretically possible except as an approximation and as to practical results, assuming the torch to perform its function perfectly, the assumption of a ratio of 1.00 ± 0.02 to 1.00 is certainly the limit of accuracy justifiable without first investigating the quantitative effect of each of the items mentioned above and correcting therefor.

Furthermore, the appearance of the flame as an indicator of gas ratio is of limited sensitivity and is a function of the light conditions, the velocity of exit of the flame gases, the absorption characteristics of the medium looked through in viewing the flame and the personal equation of the operator. These factors were not investigated by the author and while the writer is not in position to offer concrete information on each of them the combined effect is known to affect the results by at least ± 2 per cent with a tendency toward the plus error.

The results obtained in these tests give an average gas ratio for all torches in actual welding of 1.14:1 and in the flame test of 1.12:1. The best average in welding was 1.05:1 and in the flame test 1.03:1. Fig. 36 gives the results of gas-ratio tests for all torches arranged in the order of increasing ratio. These results directly contradict the statements given by the author in Par. 9*i*. Gas ratios in welding and in flame tests with few exceptions are closely related, the average difference being only 2 per cent. In view of this fact, the statement given in Par. 9*j* should apply equally as well to actual welding. Par. 9*h* is unjustified in view of Par. 9*j* the limits within which the 1:1 ratio should be expected to hold, and the fact that one or two

of the torches giving the best ratios obtained an average value of 1.05:1 in actual welding. This value is well within the limits of ratio which could be expected in view of the above items which were not taken into consideration.

The refinement of gas ratio required to produce sound welds is a factor upon which the results of this test offer little information and commercial results have shown that a ratio even as high as 1.14: 1 which was the average obtained in these tests will not pre-

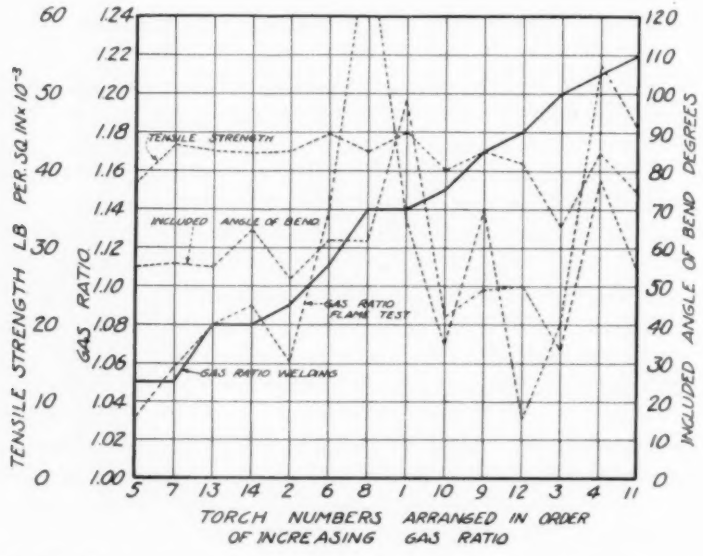


FIG. 36 RELATION OF GAS RATIO TO TENSILE STRENGTH, INCLUDES ANGLE OF BEND

vent the execution of welds having a high tensile strength and ductility when skilled operators are available.

Fig. 36 also shows tensile strength and included angle of bend of the welds. The results plotted are the results of the same tests in which the gas ratios were determined. These curves show that the tensile strength and included angle of bend depends upon factors other than gas ratio and does not justify the statement given in Par, 9*g* "that a blowpipe that cannot maintain under all operating conditions a neutral flame cannot logically be expected to produce sound welds," and in Par. 9*k* "that the question of the possible limiting strength and ductility or the efficiency of welds made by

the oxy-acetylene welding blowpipe must await the development of a more satisfactory instrument, and that having such an instrument there is no reason to believe that a weld of clean, sound metal cannot be made with assurance during any welding operation and that such welds will or can be made to possess the proper physical properties."

In order to justify these statements more elaborate tests would have to be conducted in which the personal equation is entirely eliminated, that would definitely show the extent to which the efficiency of welds is a function of the gas ratio.

In regard to the author's theoretical welding torch which would maintain an exact ratio under all conditions of restricted flow, were such an instrument possible under prescribed conditions of operations, the fundamentals of the flow of compressible fluids in orifices and tubes upon which such a condition of pressure and volume balance at the point of mixing is based also show that the change in density and viscosity due to impurities in the gases commercially available is sufficient to cause an unbalance of condition that would account for a variation in gas ratio as great as the variation from the 1:1 figure obtained with the best instruments tested. Assuming as the author does, that the orifices are of such proportions that viscosity flow does not enter into the results, the flow of gas, other things being equal, is approximately proportional to $\frac{1}{\sqrt{\text{density}}}$. A variation of 6 per cent in the acetone content of acetylene varies the density of the gas according to the following relation:

$$\text{density relative to acetylene} = \frac{\%\ \text{acetylene} \times 1 + \%\ \text{acetone} \times 2.2}{100}$$

$$= \frac{94 + 6 \times 2.2}{100} = 1.07$$

Therefore the relative flow of impure gas with the same pressure drop $= \frac{1}{\sqrt{1.07}} = 97$ per cent.

This is equivalent to a difference of pressure to obtain the same flow of 7 per cent. It is therefore obvious that such a construction which would overcome the effect of restricted flow would not be operative with gases subject to the density variation found in practice.

In view of this fact, it would be impossible to prevent the formation of a slight back pressure at the point of mixing in case of restricted flow and hence according to the author's deductions would be impossible to prevent the phenomena of "flashbacks." This

undoubtedly is true in regard to "backfiring" but the prevention of "flashbacks" or sustained backfire is dependent upon entirely different principles. The statement given in Par. 9*b* is without foundation. Many commercial torches are capable of withstanding severe abuse even with a ratio of 2:1 at the time of backfiring without causing flashback until the interiror of the torch is highly heated by repeated backfiring.

If the deductions given in this paper in regard to flashbacks were logical then the construction of cutting torches not subject to flashback troubles would be impossible without the use of a separate hose and regulator for supplying oxygen to the flame.

Despite the fact that many of the conclusions are unjustified, the data collected in this investigation are of much value to the industry and it is only through more work of this nature and even of a more searching character with the spirit of competition removed that the proper advance of the art can be made. It is to be regretted that the time and appropriations did not permit investigating some of the important phases of the problem more in detail.

THE AUTHOR. In looking over the various criticisms of the paper it was found that many of them had two or three points in common. In such cases the author has confined his reply to some one critic. Perusal of the author's replies as a whole will, he believes, furnish answers to all the criticisms received.

The author offers the following in reply to the criticisms of Mr. Banash:

The efficiency of the welds is, we believe, on the average as high if not higher than that obtained in shops where they pride themselves on producing high-class work. In many cases superior strengths reported have been produced by using special welding rods or by giving the finished weld special treatment. We are not familiar with the tests referred to but would like to call attention to the fact that there must be some mistake in the statement that the material in the tests he refers to was "practically the same," for it will be noted that the average strength of the welds as reported by him is in excess of the tensile strength of the unwelded plate used in the Bureau's tests — a fact which needs further explanation concerning his welds, especially as the ultimate strength of annealed Norway iron is approximately 40,000 lb. per sq. in. We would suggest in this connection that the best measure of the value of material used as a filler rod in welding from the standpoint of the strength of the

rod is the annealed rod and not the strength as determined on rod as received from the manufactuerer.

Relative to the blowpipe operators, the final approval of the men as based on a study of their ability was left to Mr. S. W. Miller, whom we believe will be acknowledged as fully qualified by experience to select such men; and it was Mr. Miller's statement that they were "well above he average operator in ability" and their selection was approved by him. No representative of a manufacturer who actually witnessed their work had other than complimentary statements to make of these operators. Further, one of these same operators, the cutter, was later selected by Mr. Miller as at least one of his principal cutters in an extended and important investigational test for which Mr. Miller acted as consulting engineer.

As to Mr. Banash's criticism of the cutting tests, we believe that if the fact is kept in mind that the tests were made on commercial hand cutting blowpipes, the specific statements concerning the probable maximum cut obtainable with such blowpipes need no further discussion. Relative to the question of preheating flames we still hold to the opinion that in many instances they are too large and therefore wasteful. It would seem that the statement concerning variation in acetylene consumtpion as given in Par. 62*a* would be sufficient proof of our contention. It is our belief that thoughtful observers among the users of such equipment will find that in a majority of cases no decreased efficiency of operation will result if the size of the preheating flames on some blowpipes is reduced. There is no doubt a decided tendency toward a wasteful use of gas in many cutting operations and one of the causes in many cases is in the size of the preheating flame.

Several other statements by Mr. Banash are directed toward the influence of the operator, i.e., the so-called personal equation. and the belief is expressed that on that account the full possibilities of the apparatus tested were not developed. If such a condition of affairs were true it would be of decided moment, for his argument leads to the conclusion that because of the intricacies of design of the various blowpipes it is essential that each welder be specifically taught to weld with a particular make of blowpipe and that having once been so taught, he would have to be retaught if he used another make of blowpipe. If such a thing were true it would be unfortunate from the sales point of view, especially where a manufacturer is attempting to have his apparatus displace another that is already in use. Further we doubt whether Mr. Banash will contend that

the blowpipe makes the weld; its whole office is to produce the heat necessary to make a weld, and if all blowpipes were correctly designed so as to constantly furnish the necessary heat under the proper conditions, any experienced welder could use one blowpipe equally as well as another. It would seem that the plea of "personal equation" as it affects a blowpipe is an admission of uncertainty, at least, relative to the value of design of the apparatus.

His remaining criticisms are directed to our statements concerning the quality of the test welds and a claim that our admission that the welds were unsatisfactory to us is sufficient condemnation concerning the value of the tests. It will suffice here to state that none of the manufacturers who witnessed the making of the welds (and they as a general rule had their own expert welder present during the test) or inspected them afterward had any question concerning their not being satisfactory to them. The univerasl opinion was to the effect that they were satisfactory. As to why we chose to consider them unsatisfactory we will refer to our comments on Mr. Miller's criticisms. With these things in mind we cannot agree that our frankness of statement condemns our work.

Mr. Plumley devotes considerable space to a criticism of the author's statement concerning the probable maximum cut obtainable with the ordinary commercial cutting blowpipe, and we believe there is no misunderstanding concerning the type of blowpipe reported on in these tests. He, in so doing, and in common with several others who offer criticisms on the same statement, seems to have overlooked the statement of Par. 70 of the paper. But beside that statement there is a much more interesting series of statements that it is necessary to call to Mr. Plumley's attention. The particular blowpipe Mr. Plumley refers to was purchased for test directly from the manufacturer. With the apparatus there were received a booklet of instructions and a convenient pressure card evidently intended for ready and frequent reference. In the instruction booklet there is the following statement relative to the pressures to be used for the particular cutting blowpipe under consideration: "This (the pressure necessary to produce the required cut) may be taken from the table on page 32." Page 32 of the booklet contains a table for style C cutting torches with style 12 tips. The manufacturer's shipping invoice shows that it was a style C torch with style 12 tips which were used in the Bureau's tests. The table referred to on page 32 gives tip size and pressures recommended by the manufacturer for cutting metal of various

thickness. The handy reference card mentioned above gives this table in duplicate. On this pressure table the manufacturer recommends tip number 5 with 6 lb. acetylene pressure and 150 lb. oxygen pressure for cutting metal of 10 in. thickness. While the tip size and pressures used are not reported in Table 2 because the blowpipe failed to make the 10-in. cut the records show that in three trial cuts on 10-in. thickness the number 5 tip was used with the following pressure delivered to the blowpipe at the handle valves (the manufacturers' pressures given above are regulator pressures):

	Torch handle pressure lb. per sq. in.	
	Oxygen	Acetylene
1st trial	147.8	5.88
2nd trial	147.2	5.85
3rd trial	153.6	5.86

Further, nowhere in the booklet of instruction is there any reference to larger sized tips or a suggestion that the blowpipe will not cut 10-in. material with the tip size and pressures specified by the manufactuter — any more than there is a reference that the number 5 tip is an intermediate size.

It is apparent then that the failure of the particular blowpipe in question in cutting 10-in. metal cannot be in any way attributed to any lack of knowledge on the author's part. The author has simply reported facts — facts based upon the use of the blowpipe in strict accord with the manufacturer's specific recommendations. On these facts conclusions were drawn concerning the efficiency of the type of blowpipe tested. It is the author's belief that on the facts as brought out his conclusions are correct — nevertheless it is pointed out that possibly before such conclusions can be taken as absolutely final further experimental work may be necessary. In the cases where greater depths of cut have been made, other than the regularly furnished equipment, as shown above, has been used, and in these cases nothing of which the author is aware has ever been done to determine definitely whether some other method (the oxy-hydrogen process for heavy cutting for example) is not far more economical than the use of specialized oxy-acetylene equipment.

In this connection it is desirable to point out another important condition for consideration. It is a further fact as is set forth in Par. 46 of the paper that a number of blowpipes were overrated by

their manufacturers in cutting capacity for heavy cutting. It is suggested that more conservative statements concerning such things will work for a more confident acceptance of the true value of the oxy-acetylene blowpipe.

Another of Mr. Plumley's statements is directed against the author's remarks about present-day blowpipe design and is that "The author has apparently based all his remarks and conclusions upon pressures obtained within the hose line supplying the torches." The only real use the author made of pressures within the hose lines was as a basis of classification for convenience in discussing the results. The basic statements concerning actual blowpipe pressures were founded on a set of curves of the type of Fig. 25, one of which was secured for each blowpipe tested. Incidentally the manner in which these results were obtained would make no difference in relative results had they represented pressures obtained in the head or hose of the blowpipe, and as shown by Fig. 25 for example, none of the blowpipes tested actually maintained balanced pressures within the head when any form of obstruction interfered with flow of the gas from the tip end. Further as pointed out in the paper a blowpipe might possibly possess the characteristics of balanced pressures in the blowpipe head when it is allowed to burn freely in air — but it is a fact that not any of those submitted to tests showed such characteristics when any form of obstruction to flow existed. Mr. Plumley admits in his statements concerning flashbacks that there are many common and frequent causes and opportunities for producing obstruction to free gaseous flow from the tip during the welding process.

Mr. Plumley also comments, as do several others, on the fact that a relatively large portion of the paper is devoted to the discussion of flashback. It would seem that the careful perusal of Par. 85 together with such statements as are found in Par. 86 to 91 would show the reason for the method of attack in developing the paper. As to whether we have over-emphasized the possible hazards due to the development of flashback would seem to depend upon one's view point. We suggest that Mr. Plumley read our reply to Mr. James. No matter what the method of attack the fact remains that the desideratum *sound welds* is most intimately connected with the proportions of the gaseous mixture coming from the blowpipe and therefore to the conditions causing flashbacks.

As to the possibilities of securing a perfectly balanced blowpipe, we must admit that we do not expect perfection but neverthe-

less believe in striving to attain such. The results of these tests show that the balanced blowpipe can be far more nearly attained than is at present thought possible by most manufacturers. One blowpipe in particular when fitted with one of the tips used with it in the Bureau's tests (six other similar tips when tried out did not give similar results) not only proved to be as a whole very resistant to the development of flashback and enabled the production of a group of welds of consistently high strength, but enabled the production of the best set of any of the welds made from the standpoint of the type of material deposited in the weld — practically all of it except at the bottom of the V being clean homogeneous cast metal. The dimensions in the gas-port openings in that tip were fortuitus but it showed what might be expected as to character of weld metal with properly balanced blowpipe pressures and also that a decided improvement is possible toward balanced pressure design in present day blowpipes. A critical examination of the welds made with several other blowpipes that showed high resistance to flashback development bore out the above. Incidentally these welds were made with a blowpipe the operators had never seen until it was placed in their hands for test purposes, which would seem to refute another of Mr. Plumley's points relative to the effect of the personal equation as affected by the blowpipe on the character of the weld made. Further discussion on this subject is deferred for replies to criticisms submitted by Mr. Coberly. The discussion as to the strength of welds is confined to the author's reply to Mr. Miller.

We will agree with Mr. James that ruggedness of construction, etc., may tend to minimize dangers from flashback, but at the same time we believe that he will admit and probably the records of the Underwriters' Laboratories would show that unexplained explosions etc., are more or less frequent in every large welding shop. They may not prove disastrous — more often they do not; but all such explosions must be admitted as possible causes for more serious results. For that reason a blowpipe that is capable of developing flashbacks readily cannot be condisered eminently safe — especially if such potential dangers can be still further minimized by proper design — no matter how good a risk they may be considered. The data of Table 3 which were secured before we recognized the real causes for flashbacks and which are based on duplicating tests conducted by the insurance companies show that even in blowpipes that are acceptable risks there are decidedly potential dangers. Perhaps the recognition of this fact would help

to explain the numerous small explosions occurring not infrequently in welding shops.

Relative to Mr. Coberly's comments it is true that there were many items of very minor character that the limits of the work would not permit investigating; it is very doubtful whether many of them will have the importance Mr. Coberly gives them. As regards the question of standard of quality he forgets that the investigation contemplated only determining the maximum efficiency of the blowpipes when used in accord with the manufacturer's specific recommendations. To have set a standard of quality would have required a complete investigation of each blowpipe and the resulting comparison would have been on the basis of efficiency in cutting for standard quality. To do this it would have been necessary, as indicated above, to determine the best pressure for use with each blowpipe — something we might expect the manufacturers to do — and having done that we do not see wherein one blowpipe would have shown any greater efficiency than another. In fact we should expect them to show equal efficiency then in conformity with our conclusion 9*b*. Mr. Coberly's tests set a standard of quality for one particular blowpipe fitted with a tip and operated under pressures he determined as being most efficient for it. Under these conditions he determined the data of his Fig. 34. But as we read this figure it implies that with constant oxygen consumption the speed of cutting increases with the angle of lag until the lag becomes so great that the cut fails to penetrate through the plate — an obvious phenomenon. This shows no reason, however, for the selection of a 5-degree angle of lag as a standard, except as he personally believes it gives a good quality cut. In so doing he shows by his own data that he has not necessarily selected the most economical angle. We cannot see the reason for plotting a second curve which is simply the reciprocal of the first. As for the possible error in judgment in operating the blowpipes during cutting we believe that most of the manufacturers who witnessed the test will agree that the blowpipes were speeded up to the limit of their cutting ability under the conditions specified by the manufacturer.

As for effect of impurity of oxygen — such data as are available show increased efficiency in cutting for higher degree purity and as confirmatory evidence his data of Fig. 35 are of interest. It probably will be necessary to check the relative effects of water vapor and nitrogen before his data can be taken as conclusive. The statement however that "The variation of purity during these

tests probably was sufficient to account for a variation of ± 10 per cent in the results" is in error, as all our tests were conducted on analyzed oxygen which was selected so as to maintain the purity for test purposes at 98.3 ± 0.15. The comments therefore on the preheating flames are not warranted. When he speaks here and later on of using "oxygen of high purity" we are at loss to know what he refers to. Our tests were conducted with commercial oxygen and there is no authentic evidence that we know of that shows that any of the commercial oxygens maintain a higher average purity than that we selected as standard for our tests.

Inasmuch as the oxygen and steel were stored in the same room for considerable periods of time previous to use and that our laboratory records show room temperature changes during the test period of only nominal amount, the effects of such variables may definitely be neglected. The carbon content and surface of the steel was similar for all tests. We kept barometric records, but we could not find that they were useful in determining the blowpipe efficiencies except that they were used to correct flow meter data. Under these conditions it is evident that the statement concerning our conclusions on the relative comparative merits of the various blowpipes is unjustified.

As for his criticisms on gas ratio, there seems to be no need for further comment beyond that contained in the various answers to similar remarks by other writers. But in this connection we would suggest that had he kept in mind the statements in Par. 143 to 145 inclusive of the author's paper he would not have felt warranted in commenting on our lack of investigation of the personal equation concerning the neutral flame.

We cannot see wherein Mr. Colerly's Fig. 36 refutes our conclusion 9*i*. To our mind there is no such agreement shown as he suggests by the two curves of gas ratio plotted in that figure, nor can we see on what basis he makes his comments concerning Par. 9*h* and 9*j*.

Concerning his comments on the gas ratio required to produce sound welds we will respectfully refer him to our Fig. 37 and 38 which we believe are sufficient answer as based on the question of strength. We desire to call his attention however to the fact that in judging welds he is, in common with many others giving too much emphasis to strength without considering the characteristics of the weld metal. Our comments on this on a later page will probably be of interest to him. Reference to the above mentioned

figures of course answers his further comments on his Fig. 36 and on Par. 9*g* and 9*k*.

With regard to the theoretically balanced welding blowpipe we ourselves have recognized that absolute balance probably cannot be obtained. There is no doubt in the author's mind, however, but that a far superior piece of apparatus can be developed than is now commercially available. Of the factors that will tend to cause unbalanced conditions no doubt the changing density of tanked acetylene may prove most detrimental, but is not Mr. Coberly assuming that a tanked gas changes its density by jumps when he

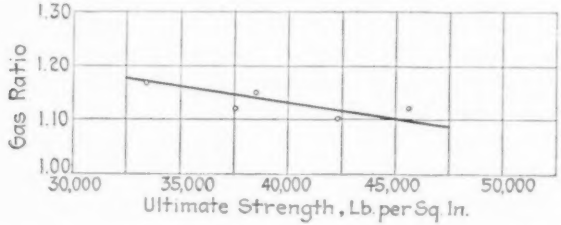


FIG. 37 VARIATION OF ULTIMATE STRENGTH WITH GAS RATIO

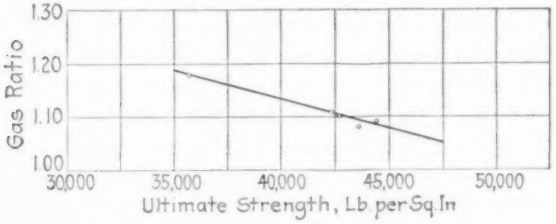


FIG. 38 VARIATION OF ULTIMATE STRENGTH WITH GAS RATIO.

makes his calculations, a condition that does not exist as far as is indicated by any data available to us? Secondly, he is basing his calculations on tanked acetylene only and in most of the larger plants we know of generator acetylene is used. Finally, he entirely neglects the possibility of purifying his gas before passing it to the blowpipe, a process that may eventually prove desirable for several reasons.

We see no reason for further remarks on his comment on Par. 9*b*, — the reason for this conclusion is fully discussed in the paper. It is unfortunate that Mr. Coberly did not offer his explanation of "flashback" since he does not agree with that given in the paper.

Finally in connection with his remarks on flashback in cutting blowpipes it is to be regretted that he apparently overlooked our Par. 9*g*. We are quite sure it will be necessary to comply with his suggestions if a true non-flashback cutting blowpipe is to be obtained. Whether this will prove necessary commercially remains to be determined — as the only drawback to a flashback in a cutting blowpipe is as a possible source of danger and not, as in welding, the production of a condition detrimental to the operation. It is purely a question here of accident risk.

Mr. Miller suggests that "some of the conslusions may be based on data that are not published." The conclusions reached were based almost entirely upon data of the type reported in the paper except that the author had at his disposal a great deal more than indicated by the paper as check or supplementary data. It is possible that in trying to keep the paper within bounds as to length, some points have not been clearly brought out. In this connection the further discussion offered now may furnish increased clearness.

In his comments Mr. Miller lays great stress upon the variation in strength of the welds as shown by the tensile and bend tests and from this draws the conclusion that they do not warrant being "used as a basis for distinguishing between the relative values of the torches tested." We ourselves called attention to this variation in Par. 81 and ended that paragraph with a statement that we thought would show the basis on which we considered the welds unsatisfactory.

Mr. Miller points out and the writer agrees thoroughly with him that the autogenous weld is a *cast weld* and that the physical properties of the weld cannot therefore be expected to be the same as those of a forged plate of even the same chemical comopsition. The first essential then is that a perfect weld should show clean homogeneous *cast* metal characteristics. Secondly, and again we agree with Mr. Miller, that a perfect weld should be one in which the cast metal should be thoroughly fused with the adjoining metal of the plate — it should be what he designates as a "sound weld". As he uses this designation, however, in a later part of his remarks, Par. 39 and 44, it would seem that he there makes it cover also the character of the fracture. If this latter comment is not true then in his suggested shop tests he is neglecting the character of the metal deposited. It would seem that right there lies the crux of the whole subject. A perfect weld or sound weld should be

1 Clean cast metal
2 Complete, or thoroughly wrought out.

The writer prefers to use the term "sound weld" to include both essentials using the term "complete" to designate its mechanical perfection. No weld can be judged satisfactory, then, unless it fulfills both essentials. A weld may therefore be complete, i.e., mechanically sound, and yet not show clean cast metal, in which case it cannot under any designation be considered a perfect or sound weld. Such a weld may even show relatively high tensile strength for though not of homogeneous clean cast metal structure it will on account of its slag and oxide enclosures be more or less fibrous and somewhat of the nature perhaps of what we term wrought iron — a strength however that may or may not be possessed by such a weld — a fact that shows the undesirability of such welds. A study of Mr. Miller's present comments leads the writer to the belief that Mr. Miller in common with a great many others is putting the most emphasis on the completeness of the weld and its tensile strength when he discusses the quality of welding, seeming to relegate to minor consideration the character of the weld metal. As to whether a perfect weld can be made, discussion will be postponed temporarily.

The analysis of the welds made by the author considered both of the essentials given above. It also took into consideration the possible personal equation of the welders. For examination purposes the edges of all test specimens were polished and etched and macrographs made of at least one sample of each test plate. We recognized the incompleteness of many of the welds, but a comparison in ultimate strength of the ten best and ten most incomplete welds as based upon macrographic selection showed a variation on averages of only 1200 lb. per sq. in.

We therefore are willing to accept Mr. Miller's statements that neither on strength of welds, completeness of welds, or operators was there apparently reason to account for blowpipe differences.

There were however several outstanding features which to the author needed explanation. Analysis showed that the tensile strengths of the welds bore a definite relation to the gas ratio (Figs. 37 and 38) and that the gas ratios were as a rule lower in the gas ratio tests (in which the blowpipe burned freely in air) than were those obtained in actual welding operations. Further, flashbacks occurred only when there was some interference with free gaseous flow from the tip. In the excess acetylene type blowpipe it was

noted that there was a constant progression and regression of an excess acetylene flame as the blowpipe was used in welding even when the pressures were maintained as nearly uniform as possible, a phenomenon which was much more noticeable when there were conditions developed that hindered free gaseous flow from the blowpipe tip. With these latter blowpipes flashbacks were seldom produced during welding, while they were quite readily produced with the excess oxygen type blowpipe. It was evident that there was some interrelation between weld strength, gas ratio and flashback and this, coupled with the fact that in the author's opinion the amount of oxide in the finished weld was in excess of what he believed there should be, considering all possible sources of such oxidization, lead to the belief that it was essential that the blowpipe designs be investigated, as no allowance for material, operators or tensile strength alone could account for such observed facts. The results of the inquiry resulting from the above mentioned conditions are given in the paper.

Mr. Miller also comments at length on the so-called gas ratio. As for the realtion between gas ratio and ultimate strengths of welds it would seem that he is selecting the exceptions rather than the general rule. Our plottings of gas ratio show that for Tests 1*a* and 1*b* there is a decided trend toward higher strengths for lower gas ratios, Figs. 37 and 38, and in Test 2 the results are too erratic to use for comparison. The curves are obtained by averaging the values given for tensile test in Table 6 and plotting them against the corresponding gas ratio given in Table 5. On the curves thus secured in order to show the trend more clearly each three succeeding points were again averaged for the new points shown in the figures. It is our belief that the erratic values for test No. 2 were due in large measure to the blowpipes themselves. The larger sized tips did not perform as well as the smaller ones; they showed greater tendencies to unevenness of flame character during welding, and by test were more subject to flashback phenomena. (This latter is shown by data of Table 3). This fact is of interest in connection with Mr. Miller's comments relative to the high values given for some gas ratios. Two figures he quotes are for blowpipe No. 4, Test 2, an excess oxygen pressure blowpipe and the most easily backflashed blowpipe tested. Our records show that during the welding operations a great deal of trouble was experienced from flashback with this blowpipe. In this selection it would look as if Mr. Miller had been of assistance in offering confirmatory

evidence. This same blowpipe had a loose ball safety valve in its rear plate, and with this design just turning the blowpipe over or shaking it changed its flame characteristics — even in gas ratios satisfactory results could not be expected of it. The values he quotes for the No. 8 blowpipe are also what might be expected, an excess oxygen pressure blowpipe with pressures for operation given at each test and without any definite knowledge on the manufacturer's part as to what results would be produced. The original pressures suggested for this blowpipe were above 20 lb. for both gases.

We cannot agree that the high gas ratios are the results of carelessness on the operator's part. Exceeding care was taken with flame adjustment and in most all cases this adjustment was approved by the manufacturer's own expert who witnessed the tests. The other possibility Mr. Miller speaks of was found to be true for several blowpipes, especially No. 9. In this latter the acetylene is admitted to the exit passage of the tip through one port hole only. The result was a stratification of the gases in passing out with the result that there was a decided tail of acetylene protruding from the inner cone of the flame. This tail on the cone could be moved from one side to the other by simply twisting the tip around in the blowpipe head and always bore a definite relation to the position of the acetylene inlet. It was necessary to use an exceedingly oxidized flame to neutralize the acetylene tail. The result of this condition is readily observed in the gas ratios obtained with grating as given in Table 5. This same trouble was experienced with the tip used in blowpipe No. 11 for test 5*a*1 made with grating except in this latter case the trouble was due apparently to the fact that the hole through the tip was drilled from both ends and the two drillings were not concentric, with the result that there was a decided shoulder that interfered with gas flow where the two drill holes met. It was the author's intention to bring this information out in his paper — it is a condition that, as can be seen, deserves attention and shows the necessity not only of proper placement of gas ports but also of careful workmanship.

When Mr. Miller asks for an explanation of a few discrepancies between values of gas ratios as given by flow meter and balance weights we believe he is again making more use of the exceptions than the general rule. In Table 5 there are seven cases of divergencies in value greater than 0.06, six of which are in values obtained in the No. 2 test or its equivalent, 5*b* — four of these latter are

in the welding test No. 2, and two of the six occur in the values given for the No. 4 torch, comment on which is made above. The recording of data for this investigation was exceedingly trying and we could hardly expect perfection in everything, — we recognize that "to err is human." Notwithstanding this, the author feels that the technical observers for this investigation rightfully deserve a high measure of credit for the continuously consistent results obtained.

The writer is not prepared to say just how much influence eyesight plays in the adjustment of the neutral flame of a blowpipe; there is no doubt it is of some importance. On the basis of Mr. Miller's figures, however, we would personally not care to attempt to say what effect it would have when we bring to mind the change in pressures that occur with the ordinary commercial regulator which he states he used.

As indicated above the necessary space limit to this paper prevented full explanations on some points. In the forthcoming Technologic Paper of the Bureau it is shown that there is considerable variation in the acetone content of the acetylene gas and that this variation depended upon several factors. It was found impracticable to carry out the determination for acetone content at this time.

Incidentally columns (7) and (8) of Table 5 show that whatever the correction for such impurities would amount to, on the final result its importance would be relatively minor.

Relative to our statement concerning comparison of gas ratios obtained in welding and in gas ratio tests our conclusion was drawn on the study of a large number of tests which it would not have been worth while to incorporate in the present paper. As indicated in the paper we carried on an extensive series of gas ratio tests; — those reported in Table 5 were the original tests of the investigation. Even these latter we feel bear out our statement. The average for the welding tests on one-half-inch and three-quarter-inch plate as compared with corresponding gas ratios for the same size tip are as follows:

Plate	Gas Ratio, Test No.		
	1*a* and 1*b*	5*a*1 (without grating)	5*a*1 (with grating)
½-in.	1.125	1.115	1.074
¾-in.	1.148	1.120	1.028

If the values for blowpipes Nos. 4, 9, and 11 are omitted (these we know are not good values on account of blowpipe peculiarities) the result will be as follows:

Plate	Gas Ratio, Test No.		
	1a and 1b	5a1 (without grating)	5a1 (with grating)
½-in.	1.113	1.097	1.029
¾-in.	1.111	1.106	1.012

Mr. Miller in his comparison was unfortunate in one respect in that in comparing 5*b* with 1*a*1 he compared the value of the gas ratio test of three-quarter-inch tips with the gas ratio value obtained in welding with the one-half-inch tips. His special comments on the grating tests are answered above in our comments on blowpipe Nos. 9 and 11 as is also his comment on the possible effect of stratification of the gases in the tip tube. We cannot agree in full however with his conclusion that "it is entirely possible to have free oxygen and acetylene in the flame, even when the gas ratio is theoretically correct." That statement needs amplification if it is not to be misleading. His statement is correct for the condition that the acetylene tail on cones of tips of the type under discussion is not completely neutralized by the oxygen in which case both free acetylene and oxygen would be present in a flame giving the proper gas ratio. If on the other hand the tail is neutralized by ejecting excess oxygen, free oxygen would be present in the flame but the theoretical gas ratio could not exist.

Mr. Miller's comments on Par. 13*g* are of decided interest. The author will not take issue with him on the possibility of distinguishing the two oxides of iron as he is not qualified by experience to discuss this. For the sake of argument we will grant that welds made by him have had equal oxidization throughout. But he states that "As far as ferrous oxide is concerned his observation is that the bottom of the weld is cleaner when the weld is sound." That means then that with the two oxides present the major portion of the ferrous oxide is in the top of the weld, that the preponderance of the oxide in the bottom of the weld is ferric oxide. Mr. Miller will of course recognize that the most oxidized of the two oxides is the ferric so that we can then say that as far as the oxides are concerned the most oxidized material is at the bottom of the weld. And are not the oxides the measure of the oxidization that has taken place? It would seem to the author that Mr. Miller has offered a further convincing proof that the bottom of the weld is the most oxidized section — a proof that is of exceeding interest as it fits the deductions of the paper exactly. He points out that there must have been a larger volume of free oxygen at the bottom of the V of the weld in order to have the ferric oxide form in preponderance.

Concerning the strength of the second weld plate over that shown by the first, we must confess we were in error in the use of a word— the sentence should read "the second plate welded generally showed higher strengths." But in making his selections is not Mr. Miller again picking the exceptions rather than the general trend, and with such regularity as shown by the figures of columns 2 and 3 of Table 6 we cannot agree that the results are fortuitous.

The question as to whether clean cast metal can be deposited consistently seems now to warrant consideration. As Mr. Miller points out, variations in the composition of the plate may have some influence on the results produced, but it is our present opinion that except for carbon content there will not be enough variation in the major elements in any similar group of good plate material to seriously affect the soundness of the weld.

As for the possible baneful effect from oxygen as it is occluded as a gas in the plate material we doubt now whether it may as yet be considered of primary importance, as to the best of our knowlege oxygen has rarely, and then only in small amounts, been extracted from ingots, etc. The common gases secured in such investigations have been N, CO, CO_2 and H, the latter generally in predominating volume. Oxidization from the plate material would probably then come from the dissociation of CO, CO_2 or the oxides of iron present in the plate, all of which are in such small proportions compared with the conditions surrounding the welding operation that their total influence as an oxidizing medium can be probably considered of minor influence.

As for the welding rod — there is no doubt that the type of rod used plays an important part in the strength of the weld resulting, but from the standpoint of the metal structure, i.e., its cast metal characteristics, or as a possible source for causing oxidation of the weld, the writer is inclined to place it in the same category as the plate. We are speaking here of course of what is generally considered good quality rod. The possible exception we would make in this is due to the experiments of the Welding Research Committee of the Emergency Fleet Corporation where it was shown that in electric welding an iron electrode that had been pickled in acid and therefore probably contained occluded hydrogen seemed to produce unsatisfactory welds. The reason for suggesting this exception lies in our knowledge of the embrittling effects of hydrogen on steel. There is a valid reason for the low-carbon rod being red short as indicated by Mr. Miller. It is known that the pure irons (decarbonized irons),

like Swedish and Armco which are free from aluminum, will forge at 700 deg. cent. (1292 deg. fahr.) or at 1200 deg. cent. (2192 deg. fahr.) but not between these temperatures. On the other hand it is possible that the three and one-half per cent nickel steel suggested will not in some cases prove any more desirable as nickel when heated over a coke fire or sulphurous fumes becomes exceedingly brittle.

It is the writer's belief that the first criterion for making sound welds lies in a corrected blowpipe design — in the use of a blowpipe that maintains balanced pressure under all operating conditions. Given that, he believes that welds of far superior quality can consistently be made by any careful welder. They may not possess very much higher strength, but their strength when the weld is complete (mechanically sound) can be relied upon. As pointed out previously an oxidized weld or a weld containing excess oxide films, slag, etc., may on account of its fibrous characteristics show relatively high strength, but such welds are not reliable and should not be so considered. If not porous and therefore increasingly subject to corroding effect they are extremely unsatisfactory for important work for, as has been repeatedly shown, such enclosures are very largely responsible for metal failures.

It is true that the products of combustion of the oxy-acetylene flame hold in themselves dangerous attributes. Carbon dioxide at temperatures above 550 deg. cent. is reduced by hydrogen. Hydrogen itself is responsible for an embrittling effect when absorbed — so is nitrogen which is present in the air as well as in liquid-air oxygen and the absorption of nitrogen is assisted by the presence of hydrogen. But these dangers will be, we believe, relatively minor compared with blowing free oxygen on molten metal as is done with present-day blowpipes.

The following figures giving the combined nitrogen in the weld metal, as determined by the Allen method, of nine samples of test welds picked at random show that the amount of nitrogen present even in welds made with an oxygen which contained nitrogen as its chief impurity is relatively low — in fact is less than 20 per cent of the amount usually found in electric arc fusion welds of steel.

Specimen Number	Combined Nitrogen, per cent
1	0.0153
2	0.0167
3	0.0206
4	0.0246
5	0.0226

6	0.0154
7	0.0254
8	0.0190
9	0.0268

Further there is also the possibility of reducing the effect of these elements by proper manipulation and it is our belief that the manipulation is most effectively expressed by Dr. Andrew McCance[1] when he says speaking on another subject but in terms that fully apply here:

"If the loss due to scaling (oxidization) is to be minimized three conditions have to be observed:

1 Melt down quickly
2 Melt down at high temperature
3 Have a low steam content in the burnt gases.

No. 1 and 2 give together hot quick working." His objection to the steam content is on account of the embrittling effect of absorbed hydrogen.

Dr. McCance's first and second principles imply, in welding work, that the metal should be melted in small amounts only at any one time and after once placed in the weld it should not again be worked over if such can be avoided. There is no doubt that with such manipulation dangers from absorption of the products of combustion, etc., will be minimized and with a properly designed blowpipe really sound welds assured.

Incidentally Dr. McCance's statement is interesting as it throws further light on the superiority of the oxy-acetylene over the oxy-hydrogen blowpipe as a welding medium for ferrous metals.

[1] Balanced Relations in Steel Manufacture, Faraday Society Transactions, 14, 1918–19, p. 214.

No. 1793

THE INTERPRETATION OF BOILER-WATER ANALYSES

By J. R. McDermet, Jeannette, Pa.
Member of the Society

The importance of taking into consideration the corrosive properties of a boiler feedwater as well as its tendency to form scale is coming to be more and more generally recognized, and the interpretation of the various types of analyses employed for the purpose forms the substance of the present paper.

Feedwater impurities fall into at least one of three groupings: dissolved solids, pollution products, and dissolved gases. The dissolved solids determinable under a technical analysis naturally group themselves into corrosive and scale-forming constituents according as they are or are not highly ionized in aqueous solutions. Pollution products are detected by inferential methods in a partial sanitary analysis, but the effects of various types of pollution other than manufacturing wastes as indicated by the analysis are concretely set forth. Dissolved-gas analyses, meaning in the main dissolved-oxygen analyses, are discussed in relation to the safe operating limits of hot-water feed lines, boilers, and cast-iron and steel-tube economizers.

An appendix translates the facts of the paper into a working plan for use in the chemical laboratory.

WITH the increasing use of economizers, higher steam pressures and higher rates of evaporation, it has become as necessary to consider the corrosive effects of a boiler water as its scale-forming properties, if not more so. Because of this the author has prepared the present paper, which discusses briefly the limitations and applications of a technical or mineral analysis; advocates the use of a partial sanitary analysis as forecasting possible trouble due to pollution; and indicates the significance of the analyses for dissolved gases in relation to the conditions under which the water is used. A laboratory procedure sheet is appended to illustrate and make useful the facts discussed.

Presented at the Spring Meeting, Chicago, Ill., May 23 to 26, 1921, of The American Society of Mechanical Engineers.

THE TECHNICAL ANALYSIS

2 The chemist in performing a technical water analysis proceeds along conventional methods and by processes of evaporation, selective solution, precipitation, etc., determines quantities which he may submit for consideration directly, calculate into terms of elemental ions and ion radicals, or hypothesize into various chemical compounds which are assumed originally to have existed in the water. Irrespective of how results are submitted, it is possible to calculate back into ion values, to which may be applied certain generalizations — with the reservations, however, that the values are arrived at initially by inflexible methods and are connected with the properties of the water only through deductive relationships.

3 The detailed instruction sheet employed to determine procedure in the laboratory and which forms an appendix to this paper, makes use of certain specific criteria. Obviously these are imperative for the automatic guidance of the analyst, and they are reproduced not as dicta—for they are arbitrary and changeable as experience accumulates—but in the hope that, being concrete values, they may lead to a discussion which will be more valuable than the paper itself. The interpretation of any type of water analysis is inferential, and the factor of experience is often more useful than chemical knowledge.

4 The dissolved chemical compounds of a water which will ultimately become scale on the boiler surface are relatively slightly soluble in a physical sense. Chemically they are characterized as feebly hydrolyzed salts, and as such they are capable of very little dissociation in water — and then only into a very limited number of ions or electrified particles; namely, those of the salt itself and those of the dissociation with water. The presence of other soluble salts affects the equilibria but little, since the initial ion combination involves an ion, forming either a weak acid or weak base. It is perfectly legitimate, therefore, to apply to this group a hypothetical combination, since the laws upon which this combination is based are at least discernible.

5 Corrosive salts, on the other hand, are characterized as being both hydrolyzed and highly ionized, one of the ions at least being that of a strong acid or base. If the chemist's ionization constants are applied to a group of these salts representative of a natural water, a bewildering series of relationships develops, due to the number of ions and their mutual effect in displacing the equilibria of each other. Such a procedure, however, is irrational for three reasons:

(*a*) Ionization constants are experimental factors determined for normal temperatures only and are not necessarily valid at boiler temperatures. (*b*) The water molecule is of the form $(H_2O)_x$ and water in its transition from liquid to superheated steam undergoes a change of molecular grouping from the dihydrol into the monohydrol form, which transition is apparently not abrupt at the change of state but gradual with rise of temperature. The monohydrol form which exists in small proportions at normal temperatures apparently occasions both the hydrolytic and dissociation phenomena. (*c*) Concentration of impurities in a boiler operating at high rating and high circulation velocity varies widely at different points in the circuit, the concentration increasing at regions where the water is abruptly accelerated. The rule of experience alone is applicable to these classes of impurities.

6 The laboratory procedure sheet employs the following basis of distinction: The values obtained in the analyses are computed into those compounds which form scale as far as the analytical values suffice. In this grouping abnormal silica is the only constituent which is capable of causing corrosion and its action is mechanical in permitting local overheating. To the remainder of the impurities limiting values are assigned, either on a percentage or an integral basis, the integral values being reserved for those constituents which are virulent. Sodium is ordinarily a fictitious quantity employed to balance combinations, but if the water analysis indicates a possibility of danger according to the criteria employed, a real value of sodium is determined and compensation is allowed for its presence. Sodium is so strongly basic in its chemical properties that its behavior in solution can be predicted with some exactness, and the hydrolyzed products of sodium salts are themselves inhibitors of corrosion.

7 It is desirable at this point, since it will be necessary later, to emphasize the fact that it is only the relatively pure waters which produce dangerous conditions, and that as the sum of dissolved solids decreases, the liability toward corrosion disproportionately increases. Beyond an impurity content of 300 parts per million, corrosive waters are almost invariably of abnormal chemical analysis so that the danger is immediately apparent. Below 50 to 60 parts per million the technical analysis, even if it shows no unfavorable features, is usually insufficient unless the history of the origin of the water is definitely known. For such waters a sanitary analysis is a distinct contribution.

THE SANITARY ANALYSIS

8 The sanitary analysis is more truly a series of tests in which the reaction of the water is observed as indicative of the pollution which it has undergone, than a determination of foreign constituents. Technique has been so highly developed by the sanitary chemist that it is possible for him to tell from the values obtained in analyses the extent of the pollution and its sources — whether they be industrial, organic vegetable, or wastes from animal or human sources.

9 Industrial pollution usually manifests itself in the form of some characteristic compound which is a by-product of processes of manufacture. Aside from mine and paper-mill wastes, which usually reveal themselves unmistakably in the technical analysis, this class of pollution may occur in extreme dilution and yet be harmful in pure waters. The detection usually arises in abnormalities in the sanitary analysis, but the isolation of the distinct cause is often a matter of tedious investigation. Usually it is sufficient to search for the source, knowing that pollution exists, rather than to work from the specific pollution backward.

10 Under the heading of organic vegetable contamination may be grouped the products of algal growths, water plants in stagnant bodies of water, and the run-off from uncultivated land, particularly marshes, and deforested or forest-fire areas. Chemically such contamination manifests itself either as acidity due to weak organic acids, or as nitrogen compounds. In the power plant the physical damage resulting from organic acids appears principally in the corrosion of saturated-steam lines, turbines, and vapor spaces of condensers; in other words, in apparatus where moisture accumulates from partial condensation. Unlike dissolved-gas contamination, the feed lines and boilers where the dilution of the acids is very great are reasonably exempt, and the corrosion does not exhibit its greatest vigor in a narrow zone of temperature. Undoubtedly the presence of dissolved oxygen accelerates the corrosion, but the organic acids polymerize or change their form rapidly and the action is not indefinitely cyclic as in the case of some mineral acids.

11 Nitrogen compounds — and in particular ammonia — originating in organic vegetable pollution confine their attacks principally to copper and brass fittings and the iron is exempt in the path of steam flow. In dead-end spaces serious corrosion may occur on both ferrous and non-ferrous metals, but aside from this the damage is usually inconsequential. Dissolved carbon dioxide in

appreciable concentration due to bicarbonate alkalinity is an accelerator.

12 Sewage contamination may be classified as recent or remote, and further divided into household refuse and animal or human excrementitious matter. The latter, unless it contributes a dangerous chlorine content, sludges in the boiler and may be dismissed without further comment. Oleaginous matter and greases from household wastes if of recent origin contribute to foaming, priming and corrosion much as do organic acids if in considerable concentration. Sewage of remote origin which has been oxidized by aeration and bacteria action shows high nitrate content and sometimes nitrites, depending upon the extent of the natural purification. While the criteria in such cases are the nitrate content and the qualitative presence of nitrites, the introduction of both of these into a boiler water in any concentration is earnestly to be avoided.

THE DISSOLVED-GAS ANALYSIS

13 The analysis for dissolved gases is confined entirely to the determination of dissolved oxygen. The position of dissolved carbon dioxide is anomalous. The researches of the author have been concerned with the simultaneous removal of carbon dioxide and oxygen, and are inconclusive; the published results of Speller [1] incline to the view that carbon dioxide in the absence of oxygen is not prejudicial. In European practice carbon dioxide is regarded with as much distrust as oxygen, due probably to its theoretical tendency to polymerize into acids of the formic-glycollic series. Cases where corrosion directly chargeable to carbon dioxide has occurred have usually accompanied a high-bicarbonate alkalinity or been complicated by ammonium compounds. It is probable, however, that increased rates of driving will compel an entire reversion of attitude toward boiler-water alkalinities.

14 Analytical methods for determining dissolved oxygen are available in standard chemical textbooks, and methods of manipulation adapted to the engineer's needs have been discussed in the technical press.[2] It is generally impossible to transport samples of

[1] A Method for Practical Elimination of Corrosion in Hot Water Supply Pipe, F. N. Speller, *Jour. Am. Soc. Heating and Ventilating Engrs.*, vol. 23, no. 9, January 1917.

[2] Simple Methods for Determining Dissolved Gas Content of Boiler Feed Water, J. R. McDermet and D. Wertheimer, *Power*, Nov. 2, 1920, p. 686.

water drawn from significant places in the feedwater cycle to the laboratory for this type of analysis, on account of vitiation. A severe limitation is therefore placed upon the method employed, in that it must be embodied in apparatus which can be taken into the power plant and used there. This feature of portability has made chemical methods more adaptable for the purpose. The Winkler iodiometric titration method is desirable in all cases where the water is free from nitrates or nitrites. For such waters Lenossier's absorption by a calibrated solution of ferrous sulphate is best suited.

15 Methods for determining dissolved oxygen fortunately and unlike other analyses on water determine definitely and directly the constituent sought, and it only remains to apply the information intelligently to the elimination of corrosion problems. While deterioration of the ferrous metals results at all stages of the water and steam cycle where water and oxygen are present as such, in all but two cases which have developed in the writer's experience alleviation has been sought only for steel-tube or cast-iron economizers, boilers, or hot-water feed lines. The characteristic manifestation is the wasting of cast iron and pitting of steel surfaces, sometimes accompanied in the latter case by a pseudo-stalagmite formation.

16 It is impossible with commercial pipe materials to formulate an opinion as to how long a hot-water boiler-feed pipe system will last. It appears, however, if the oxygen content is kept below 0.7 cc. per liter (0.7 part in 1000 by volume), the gas volume being reckoned under standard conditions, the life of the feed lines will be of such a length as to prevent ascribing the ultimate failure to oxygen. It may be taken as axiomatic, however, that the reduction of oxygen content to the lowest consistent value is always a desirable precaution.

17 It is impossible to set any limit on dissolved-gas content in the water fed to boilers; too much depends on the conditions of boiler operation, which must be definitely known before an opinion can be offered. It is possible, however, to offer some generalizations which must be supplemented by minutiæ in actual practice.

18 Boilers that operate below rating rarely suffer any damage, whatever may be the oxygen content of the feedwater. As the rating goes up, however, damage appears, but if the concentration is kept below 1.0 cc. per liter the boiler seldom experiences pitting action. Boiler scale is a fairly efficient protective agent against this form of attack. Unfortunately, scales of predominant silicate or pure sulphate composition introduce a different corrosion loss, in

that there is an amalgamation at the boundary between the iron surface and the scale through the intermediation of iron oxide. The use of mechanical cleaners in removing this scale results in a severe detrition loss from the iron surfaces which may very markedly shorten the life of the boiler. The removal of oxygen from the boiler feed to the extent that existing apparatus is able to accomplish it, does away with this amalgamating action, and these classes of impurities in the absence of heavy carbonate formations become self-scaling.

19 Boilers rarely present simple corrosion phenomena: usually trouble arises from a complexity of causes. Corrosive dissolved solids which yield free acid radicals and dissociated acid radicals and dissolved oxygen are mutually accelerants of pitting action. With such boiler waters the removal of dissolved oxygen is a worth-while alleviation and a guarantee against regenerative action with a stable acid ion, but it is not a cure. Chemical treatment of the water is the most logical corrective if it is properly controlled and is not based on excessive alkalinities; but in practice the results obtained are usually far short of the ideal and need to be supplemented. The pitting of the surfaces presents the same general appearance. Where it is directly chargeable to dissolved gases the action is more or less selective, in that the area of greatest virulence is confined to parts where the circulation is sluggish, and entirely to the water space. In boilers subjected to severe acid corrosion, however, the pitting may extend into the steam space.

20 The corrosion of a cast-iron economizer likewise presents two aspects. The corrosion of cast iron in its initial stages is more rapid than that of steel, but the adhering residue products from the corrosion — graphitic carbon and silica — immediately form a protective coating which arrests further action. Cast-iron economizer corrosion is due to the continuous removal of the protective coating, has most nearly the appearance of erosion, and takes place in the headers and tube ends where water velocities are changed. This erosive-corrosive action will disappear if the oxygen content of the feedwater is kept below 0.7 cc. per liter, provided that the water does not deposit scale on the economizer surfaces. Economizers in the past have been principally waste-heat adjuncts operating between the extreme temperature limits 90–250 deg. fahr., and if these limits are extended in future practice it will doubtless be necessary to lower further the oxygen value. Where the temperature and character of water combine to produce scaling on the cast-iron

tubes which necessitates the periodical use of mechanical tube cleaners, the safe oxygen limit is automatically reduced to 0.3 cc. per liter. The tube cleaner not only removes the existing protective coating which has formed, but disturbs the surface layer so that a new protective coating is slow in forming.

21 Jacobus and Speller [1] independently and by different methods have established the value 0.2 cc. as the maximum oxygen content which will prevent accelerated corrosion in a steel-tube economizer. The allowable content which will permit the realization of a profitable operating life from the economizer may conceivably be lower than 0.2 cc. per liter, but this can be determined only after years of experience.

22 The most virulent steel-feed-pipe corrosion occurs between the temperature limits 180–200 deg. fahr.; no such range is exhibited by the steel-tube economizer. Water in its passage through and temperature elevation in the economizer gradually rejects a portion of its dissolved gases in the form of bubbles which cling to the tube wall and by coalescence establish for themselves a slower rate of travel than the water, whose velocity is materially below the region of turbulent flow. Besides materially reducing conductivity, these bubbles through their oxygen content both occasion the corrosion and distribute it over the entire surface.

23 The characteristic corrosion phenomena in the case of steel-tube economizers are pit holes. The primary cause for these may be exceedingly small pieces of imbedded mill scale or segregated impurities in the metal which form electric cells and occasion roughness, or an initial roughness on the surface, to which the bubbles of gas adhere and produce minute air pockets. The appearance of a pit hole is accompanied by a stalagmite or barnacle-like growth of corrosion products completely covering the pit hole and projecting above the surface. These barnacle-like protuberances consist of three successive layers of iron hydroxides in the order of their proximity to the iron surface, $Fe(OH)_2$, $Fe_3(OH)_8$, and $Fe(OH)_3$, which in their dehydrated forms correspond respectively to the ferrous, black, and red oxides of iron. The hydroxide $Fe_3(OH)_8$ only is adherent, and constitutes the binder and the bulk of the volume. These hydroxides possess different electric potentials as between each other and the iron surface, and the transfer of oxygen is apparently be-

[1] Discussion by F. N. Speller of paper on The Separation of Dissolved Gases from Water, by J. R. McDermet, presented at the A.S.M.E. Spring Meeting, St. Louis, Mo., May 24–27, 1920.

tween the superimposed layers of corrosion products and the final iron surface. The growth of the projecting surfaces further arrests the bubbles and the electrolytic potentials accelerate the action, but all causes combine to keep it very localized. The result is a punctured surface.

APPENDIX

LABORATORY PROCEDURE IN WATER ANALYSIS

(1) If the water originates with the Condenser Division as a water for circulation, perform (7), calculate according to (8) and report on this basis.

(2) If the sample originates with the Air Separation or Research Divisions, perform the analysis according to the detailed procedure which follows. A memorandum from the chief of the division concerned will be issued covering as far as data permit the following facts:

- *a* Purpose of the analysis
- *b* History and origin of the sample
- *c* Any physical observations which would direct a preponderance of suspicion toward peculiarities to be expected in the analysis.

(3) Record all numerical results in parts per million (p.p.m.).

(4) Record from observation in clean glassware qualitatively —

- *a* Turbidity
- *b* Presence of visible suspended matter
- *c* Color
- *d* Odor hot
- *e* Odor cold.

(5) Determine by titration —

- *a* Acidity or
- *b* Alkalinity to phenolpthalein
- *c* Alkalinity to methyl orange.

(6) Calculate from (5) —

- *a* Acidity as minus CO_3
- *b* Alkalinity as plus CO_3
- *c* Causticity as (OH).

(7) Analyze and calculate —

- *a* Suspended matter
- *b* Total dissolved solids
- *c* Loss by ignition (indicate qualitatively if there is any blackening of the residue during heating)
- *d* Inorganic dissolved solids
- *e* Calcium
- *f* Magnesium
- *g* Aluminum
- *h* Iron and manganese as iron
- *i* Silica
- *f* Sulphates

k Chlorides
l Nitrates (nitrometer)
m Ammonium (free)
n Bicarbonates (H CO_3)
o Carbonates (CO_3)
p Hydroxides (OH)
q Calculated inorganic dissolved solids (sum of (5-*e*) to (5-*p*) inclusive).

(8) When hypothetical combinations are required, calculate as follows and make out a careful balance sheet:

a Iron, manganese, aluminum as sulphates
b Calcium as carbonate
c Any remaining calcium as sulphate
d Any remaining calcium as nitrate
e Any remaining calcium as chloride
f If any carbonate remains, calculate as much of the available magnesium to carbonate as possible
g Any remaining magnesium to sulphate
h Any remaining magnesium to nitrate
i Any remaining magnesium to chloride
j Any remaining anions to corresponding sodium compounds, the value for sodium being assumed
k Express silica as such.

(9) Apply to (7) and (8) the following criteria:

a If the silica exceeds 5 parts per million (p.p.m.) and the causticity 30 p.p.m., report as industrial pollution and discontinue. If silica exceeds 5 p.p.m. and the causticity is low, report as having passed through sanitary filtration for verification.
b If the calculated inorganic dissolved solids (7-*q*) are less than 60 p.p.m. or acidity is indicated under (5-*a*), determine acidity or alkalinity by an electrometric titration and report as (9-*b*).
c If the calculated inorganic dissolved solids (7-*q*) lie between the limits 60–300 and the sum of (8-*b*), (8-*c*), and (8-*k*) is less than 60 per cent of this total, or if chlorides (7-*k*) exceed 25 p.p.m. or nitrates (7-*l*) exceed 10 p.p.m., determine a real value of sodium-potassium and report as sodium under (9-*c*).
d If the calculated inorganic dissolved solids (7-*q*) exceed 300 p.p.m., discontinue.
e If the sum of the calculated inorganic dissolved solids (7-*q*) is less than 60 p.p.m., apply the sanitary analysis which follows.

(10) Determine —

a Oxygen consumed by permanganate on two samples within the temperature range 60–80 deg. fahr., one at the end of 15 minutes, one at the end of 3 hours.
b Nitrites (qualitatively)
c Nitrates (transfer (7-*l*))
d Free ammonia
e Albuminoid ammonia
f Kjeldahl nitrogen.

(11) Interpret the limits as follows:

a Oxygen consumed: High = 4 p.p.m.; low = 0.6 p.p.m.
b Nitrites (qualitatively): High, distinct, faint, very faint trace, none.
c Free ammonia: High, 1 p.p.m.; low, 0.1 p.p.m.
d Albuminoid ammonia: High, 0.2 p.p.m.; low, 0.02 p.p.m.
e Nitrates: High, 10 p.p.m.
f Chlorides: High, 25 p.p.m.

(12) Apply the limits of (11):

	Free ammonia	Albuminoid	Chlorine	Inference
a	High	Moderate	Low	Sewer gas
b	High	High	High	Sewage
c	High	Low	High	Human pollution
d	Moderate	Low	Very low	Vegetable matter

e High oxygen consumed, particularly in the presence of low acidity, is confirmatory of sewage.

If the albuminoid nitrogen is approximately one-half the Kjeldahl nitrogen, it is confirmatory of high organic purity. If it is appreciably greater than one-half, it indicates pollution.

DISCUSSION

D. S. JACOBUS. The author refers to the maximum oxygen content that cannot be exceeded in the feedwater without causing an undue amount of corrosion in a wrought-steel economizer, which Mr. Speller and the writer set sometime ago at 0.2 cc. per liter.

The feedwater in the average plant contains from, say, 2 to 4 cc. of oxygen per liter and by reducing the oxygen to 0.2 cc. per liter the greater part is expelled. There will be some corrosion, however, with an oxygen content of 0.2 cc. per liter and it is well to reduce the oxygen considerably below this in order that the amount left in the water may have but little or no corrosive effect.

The air can be removed from the feedwater to the above extent by heating it in an open heater to about the boiling point at atmospheric pressure. When this is done, considerable of the economizer effect is lost through the lowering of the temperature range between the hot gases and the economizer, as it is, of course, advantageous from the standpoint of efficiency to feed the water to the economizer at as low a temperature as possible. There is also a loss on account of the vapor which escapes through the vent pipe of the heater. The additional efficiency secured in the economizer through feeding the water at the lowest temperature that can be carried under practical operating conditions without involving difficulty through sweating, say 120 to 140 deg. fahr., as compared with feeding it at a temperature approaching 212 deg. fahr. varies with the boiler and

economizer arrangement, but a fair average is about one and one-half per cent. To this must be added the loss at the heater when the water is heated to nearly the boiling point, which may be as great as the loss at the economizer, or more.

The author has not brought out the fact that he has been working with the Elliott apparatus for the removal of air at ordinary feedwater temperatures so as to enable the gains just mentioned to be accomplished. In the Elliott apparatus the water is heated to a temperature somewhat higher than that at which the water freed from air enters the economizer. It is then sprayed into a separating chamber which is maintained under a partial vacuum and an explosive or sudden boiling takes place which liberates the air. The air, together with the water vapor, is removed by means of a steam ejector. The steam from the ejector and the water vapor produced by the boiling are condensed in a small condenser cooled by the feedwater and afterwards returned to the heater. The air is vented off from the system at the small condenser. The water for the economizer is taken from the bottom of the separating chamber. No heat is lost from the system other than the amount that is radiated.

Aside from overcoming the effect of the oxygen in causing corrosion, it is advantageous to remove the air and contained oxygen from the feedwater before it is fed into the economizer and boiler as this enables a higher vacuum to be maintained in the main condenser of the power plant, which in itself results in an increase in the plant efficiency. This action is ordinarily a small one and it may be obscured through air leakage at the condenser, but it nevertheless has some effect in increasing the efficiency.

The use of economizers in this country is growing on account of the movement to obtain higher plant efficiencies to offset the increased cost of fuel. With the increase of working pressures, wrought-steel economizers are coming more and more into use, and in order to prevent an undue amount of corrosion a study of the feedwater conditions must be made along the lines brought out in the author's paper. The ideas of most of us on the treating of boiler feedwater are old, and the author should be commended for making us realize this and causing us to put some new thought upon the subject.

G. E. Pfisterer. In cast-iron economizers corrosion due to oxygen or CO_2 may be assumed to be almost a negligible quantity.

Where the temperature range of the water is 120 to 250 deg. fahr. the precipitate thrown down can be washed out with ordinary water pressures of 50 lb., and unless the water contains an abundance of carbonate, the scale formation under the above temperature range is not great. Tube cleaners may disturb the inner surface of the tubes, particularly if the cleaners are not properly set and work to limits which allow them to disturb the metal part of the tube itself.

In steel economizers the pit holes referred to by the author are not unusual. The cause may be as mentioned, and the result is a punctured tube. The writer has seen the pitting mentioned; in one case on a preheater the stalagmites were as large as peas and the heater was in service only three months after a complete overhauling, when it showed corrosive effects sufficient for it to be taken off the line.

It is the writer's impression that considerable experience may be gained from the various installations of steel and cast-iron economizers now in use as regards the corrosive effect of oxygen and CO_2. Until there is some definite data regarding the maintenance of both types of machines, it will be difficult to make a comparison. There are figures now on cast-iron economizers and we await with interest data on the upkeep, maintenance and depreciation of steel economizers.

Charles Russ Richards. When they are present in boiler waters, sodium carbonate and ammonium carbonate may frequently cause considerable trouble. Under certain conditions, sodium carbonate may be changed to sodium hydroxide within the boiler, and the presence of this material may cause the boiler plates and tubes to become so brittle as to be unsafe. The University of Illinois has had to contend with this difficulty. After three winters' use, two 500-hp. Babcock and Wilcox boilers were so badly affected that all of the drums and many of the tubes had to be replaced Relief from this trouble has been secured by treating all boiler "make-up" water with sulphuric acid. The presence of ammonium carbonate in boiler water may lead to the liberation of free ammonia which will attack the brass and copper parts of valves, etc. We have found it necessary to substitute steel diaphragms in our radiator regulating valves because of the rapid destruction of the copper diaphragms with which such valves are normally equipped.

No. 1794

THE HYDRAUCONE REGAINER, ITS DEVELOPMENT AND APPLICATIONS IN HYDRO-ELECTRIC PLANTS

By W. M. White, Milwaukee, Wis.
Member of the Society

Those familiar with hydroelectric power-house operation where curved draft tubes are employed cannot have failed to notice the boiling and disturbance of the water at the outlet from the curved tubes, which disturbance is clearly an evidence of inefficiency and loss of energy. The device described in this paper — the hydraucone regainer — provides a means for efficiently recovering the energy discharged from the runner for useful effect on the water wheel within the limited space available in the power-house foundations.

This new method of regaining pressure from velocity of fluids in motion consists in causing the stream flow to impinge upon some definite shape, either flat, conical or concave, thus changing its direction, and then placing an envelope around this shape so formed upon the particular base used, which envelope conforms to the shape of the fluid at entrance and gradually recedes from what would be the normal or free shape of the non-enclosed fluid impinging upon the particular base used; the effect of this gradually diverging envelope being to change the velocity head of fluids flowing at high velocity into its entrance into pressure and low velocity at its exit.

The field of application of the hydraucone regainer, according to the author, will be greater on low-head plants, although it is now being installed in connection with two 40,000-hp. units which are to operate under a head of 421 ft. Hydraucone regainers are now in operation or in course of construction in a number of plants to operate under heads varying from 8 ft. to 421 ft. and with waterwheels developing from 150 to 40,000 hp.

The paper gives very complete particulars of the long series of experiments resulting in the development of the hydraucone regainer, as well as data of a series of tests made at the Holyoke Testing Flume showing the increase in power-plant efficiency obtained by its use.

"HYDRAUCONE" is a new word coined for convenience in referring to the new method of transforming velocity head of fluids in motion into pressure head. In coining the word the letters "hydr" were intended to express a relation to the field of hydraulics, "au" to convey the impression of water in motion, and "cone" because of its general shape.

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2 The new method of regaining pressure from velocity of fluids in motion consists in causing the stream flow to impinge upon some definite shape, either flat, conical or concave, thus changing its direction, and then placing an envelope around the shape so formed upon the particular base used, which envelope conforms to the shape of the fluid at entrance and gradually recedes from what would be the normal or free shape of the non-enclosed fluid impinging upon the particular base used; the effect of this gradually diverging envelope being to change the velocity head of fluids flowing at high velocity into its entrance into pressure and low velocity at its exit. The angle of divergence of the envelope from the normal or free shape of the impinging fluid is such as to bring about the same phenomena of flow condition as result in an expanding straight-axis tube such as the discharge end of a venturi meter. The new method therefore priovides a means for transforming velocity head into pressure head wthin a short distance measured along the axis of the inflowing stream, and has useful application in water-power plants, supplanting the usual curved draft tube and providing for higher efficiencies with a minimum of excavation. For clearness reference is only made in this discussion to the single-Francis-runner vertical water wheel unless otherwise stated.

3 The draft tube of a water wheel serves two purposes. The first of these is to maintain at the discharge side of the runner passages a suction action equivalent to the difference in elevation between the runner and the level of the tail water, thereby providing for the use of the total head. The second purpose is to transform the velocity head in the water as it is discharged from the runner into pressure head by the time the water reaches the end of the draft tube, and thus maintain at the discharge side of the runner passages a suction action greater than that which would be caused by the difference in elevation between the runner and the level of the tail water.

4 At first thought it seems like attempting to lift oneself with one's bootstraps to endeavor to utilize the energy in water after it has left the water-wheel runner, but this utilization may be effected by an efficient transformation of the velocity head in the water discharged from the runner into pressure head, which causes an increased suction action at the runner and thus maintains a greater head acting on the runner than the total head as measured between the equivalent water level of the water at entrance to the water wheel and the level of the water in the tail race.

5 The necessity for an efficient means for regaining pressure

from velocity of the water discharged from the water-wheel runner in modern power plants has been brought about by the ever-increasing demand for higher speeds of water wheels for given conditions of power and head. The cost of electric generators of such sizes and speeds as are usually employed in water-power plants varies inversely as the speed, hence the buyer maintains a constant urge for higher water-wheel speeds, to meet which many high-speed types of runners have been brought out. As the specific speed of a runner increases, the percentage of energy it discharges also increases. A runner of moderate specific speed discharges water with such velocity that the energy contained in the water as velocity head is about 10 per cent of the energy of the total head; and runners of highest specific speeds discharge the water at such velocity that the energy contained in it as velocity head is from 20 per cent to 25 per cent of the energy due to the total head acting on the water wheel. In order to secure high efficiency with high-speed runners it is therefore vital to regain the major portion of the energy contained in the water discharged from the runner for useful effect upon the water wheel, for as more of the energy which is discharged from the runner is regained, the efficiency of the water-power plant is increased.

6 A good illustration of the necessity for a better means of regaining the energy discharged from water-wheel runners is afforded by the installation at the Aluminum Company of America's plant at Massena, New York, where five 6000-hp. vertical two-runner water-wheel units are installed. Efficiency was of great importance, and the contract was placed with the manufacturer upon a bonus and penalty basis for efficiency, this being $6000 for each per cent of efficiency per unit. The contract value of 1 per cent increase in efficiency in this development was therefore $30,000. Very elaborate tests were conducted by the Aluminum Company of America to determine the efficiency of these water wheels, the final results of which showed that the guaranteed efficiency was exceeded by about 2 per cent, yet the efficiency could have been increased very much more had a better form of draft tube been installed.

EXPERIMENTAL WORK RESULTING IN THE DEVELOPMENT OF THE HYDRAUCONE REGAINER

7 The necessity for a more efficient means of regaining pressure from velocity and changing the direction of flow from the water wheel within the limited space available in power-house foundations led the

writer to undertake a long series of experiments resulting in the development of the hydraucone regainer as set forth in the following pages.

8 A hydraulic laboratory was installed in a garage near the cornor of Second and Wright Streets, Milwaukee, Fig. 1 showing a view looking into the main doorway and Fig. 2 a view of the water

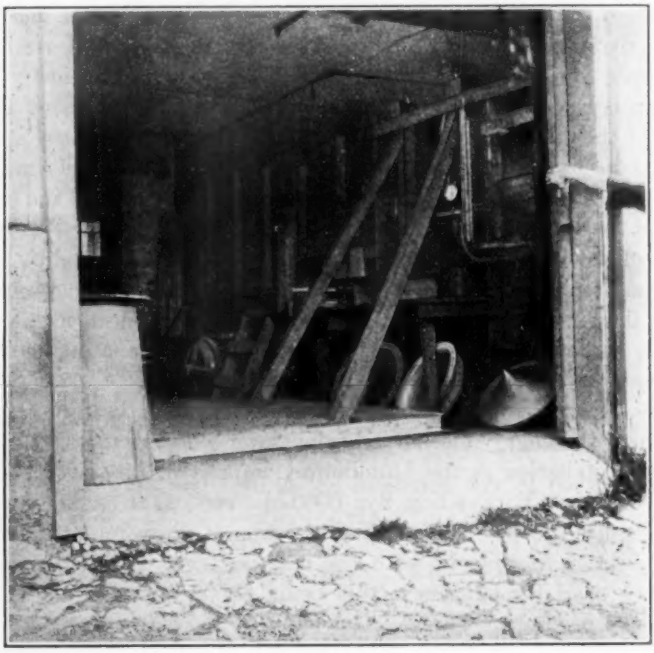


FIG. 1 VIEW OF LABORATORY THROUGH MAIN DOORWAY

wheel and of the water box. The essential equipment of the laboratory was as follows:

A water box 6 ft. wide, 5 ft. 6 in. high, and 32 ft. long

A centrifugal pump of 14 in. discharge diameter, belt-driven by a 25-hp. motor which served to supply a maximum of 14 cu. ft. per sec. of water against a maximum head of about 15 ft.

A supply pipe leading from the centrifugal pump to a 42-in. diameter cylindrical plate-steel water-wheel casing with side inlet

A water wheel equipped with wicket guide vanes

A high-speed runner (see Fig. 3), being a 9-in. model of type No. 13, Allis-Chalmers Mfg. Co. The discharge diameter of the runner where it joins the draft tube is $13\frac{3}{8}$ in. Specific speed, 90 (English system)

A prony brake
A tachometer
Necessary water gages and manometers for measuring head of water
A weir 6 ft. wide without end contractions, and having a machined brass weir notch
The necessary baffles for smoothing the water flowing from the water wheel to the weir
A collecting chamber and suction pipe leading to the centrifugal pump
Various pitot tubes, manometers and gages for determining velocities and pressures in, around, and about the hydraucone regainers and curved draft tubes.

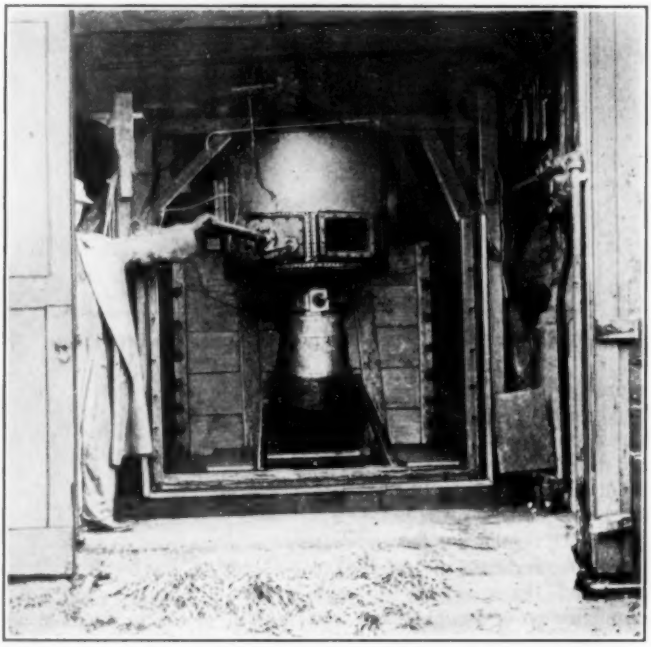


FIG. 2 VIEW OF WATER WHEEL AND WATER BOX IN LABORATORY

9 The end of the water box was made adjustable so that the curved draft tubes could be tested, as shown in Fig. 2. Tests of the hydraucone regainer were made by using the full length of the box and submerging the hydraucone regainers.

10 The output of the water wheel under good testing conditions was about 8 hp., so that the size of the equipment was sufficient to give a practical demonstration of the relative efficiencies of draft tubes, hydraucone regainers, and other devices tested.

11 A great many tests were made with many objects in view, from which the following are selected for discussion as giving a fair record of the comparative efficiencies of draft tubes and hydraucone regainers:



FIG. 3 VIEW OF RUNNER USED IN LABORATORY TESTS

No draft tube
Straight conical draft tubes Nos. 1 and 2
Short curved draft tube
Argo draft tubes Nos. 1 and 2
Mio draft tube
Steel plate curved draft tube
Hydraucone regainers with cone centers
Hydraucone regainers with conical plates
Hydraucone regainers with flat plates
A combination of straight draft tube and hydraucone regainer with flat plate.
Power-house model of hydraucone regainer.

The hydraucone-regainer tests were varied by using conoidal chambers of different shapes.

12 The actual test work was done by Messrs. J. J. Ring, B. S., and Edward W. Burbank, B. S., Assoc-Mem. Am. Soc. M. E. Work on this series of experiments began in the fall of 1913 and continued until the summer of 1916.

13 The test with no draft tube was made by removing the curved draft tube shown in Fig. 2, replacing the end of the water box,

and bringing the water up around the bottom of the water wheel and casing. The bottom of the runner is about flush with the bottom of the water-wheel casing. The water discharges from the runner at best efficiency with a decided forward whirl which rotates the water underneath the casing, and this in a measure may be responsible for the higher efficiency of the runner without draft tube than when provided with a short curved draft tube within which the whirling velocity of the water cannot be regained.

14 The straight conical draft tubes shown in Fig. 4 are the

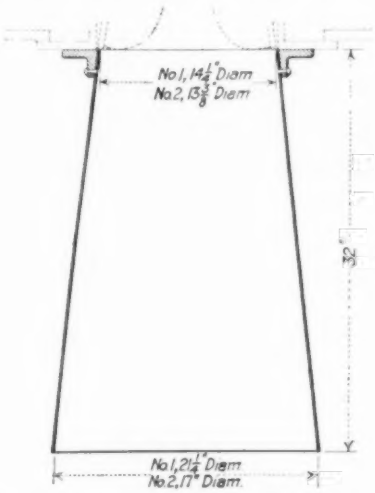


FIG. 4 STRAIGHT CONICAL DRAFT TUBE NOS. 1 AND 2

longest straight draft tubes that could be installed in the water box (Fig. 2). The dimensions are shown in the figure.

15 Dimensions of the short curved draft tube are shown in Fig. 5. This draft tube is a model of one of the shortest curved draft tubes in use in modern water-power plants. The computed-area curve from which the intended deceleration of velocity can readily be determined, is shown at the right of the figure.

16 Fig. 6 gives dimensions of a model of one of two draft tubes used with two identical water wheels installed at the Argo Plant of the Eastern Michigan Edison Company. The dimensions and shapes of the two tubes do not differ very much, and the tests at the laboratory showed that the performances with the model water wheel are practically identical.

17 The dimensions of the model draft tube shown in Fig. 7 are proportional to the draft tubes constructed in concrete at the Mio Plant of the Consumers' Power Company of Michigan.

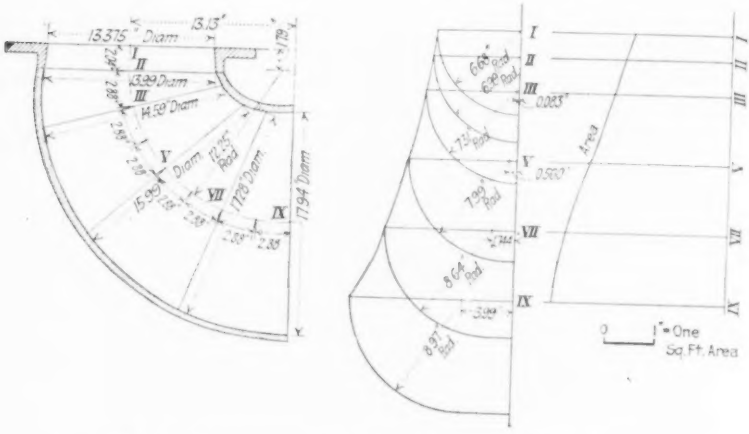


FIG. 5 DIMENSIONS OF SHORT CURVED DRAFT TUBE

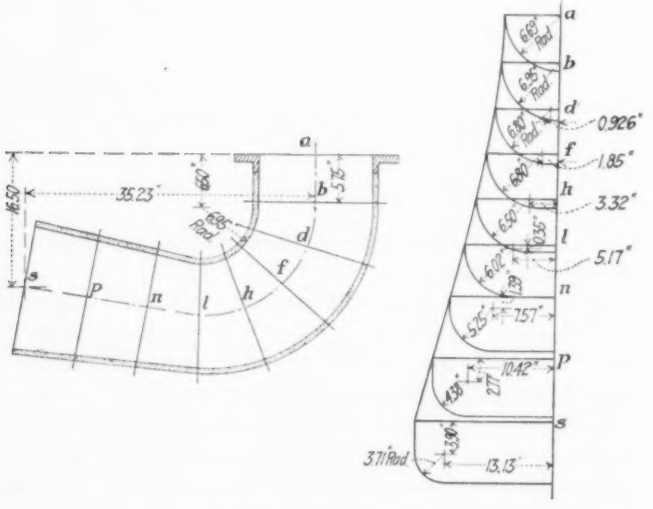


FIG. 6 DIMENSIONS OF MODEL OF ARGO CURVED DRAFT TUBE

18 The hydraucone regainers with cone centers are shown at *A* and *B* in Fig. 8, and the hydraucone regainer with conical plates at

C of the same figure. The hydraucone regainers with flat plates are shown in Fig. 9. Figs. 9*a* and 9*b* are illustrations of the cones shown in Fig. 8.

19 Fig. 10 shows a combination of a straight draft tube with a hydraucone regainer having flat plates. This type, now frequently employed, was the one installed at Niagara Falls, as will be more

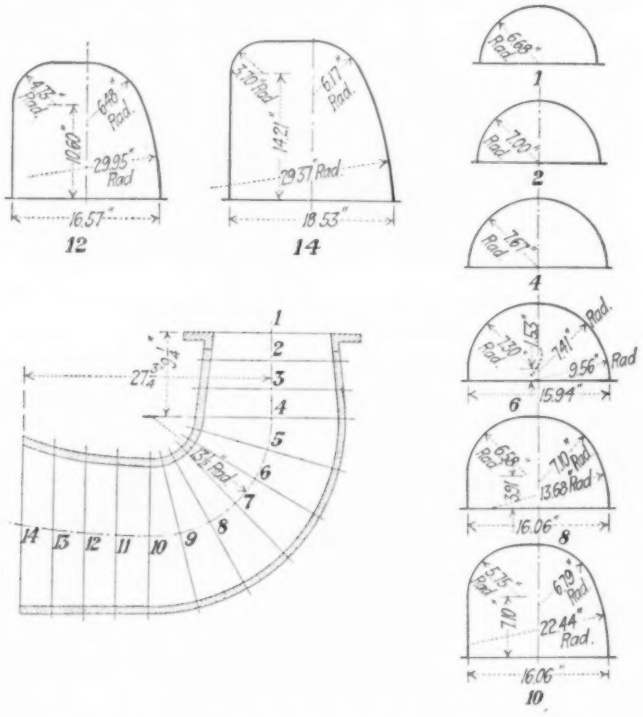


Fig. 7 Dimensions of Model of Mio Draft Tube

fully described later. Fig. 11 gives the dimensions of a model of hydraucone regainer such as has been installed in several power plants now in operation.

20 Figs. 12 and 13 show respectively the test curves plotted from the results of data obtained in the tests of the models of the Argo draft tube No. 1 and the hydraucone regainer with flat plate. It will be noted that the test results were all reduced to a basis of 1 ft. head before the results were plotted. Examination of these curves

should convince any one that the tests must have been carefully conducted, and that the results may be relied upon to give at least a fair comparison of the relative efficiencies of the various devices tested. The identical water wheel, equipment and setting were used in all these tests.

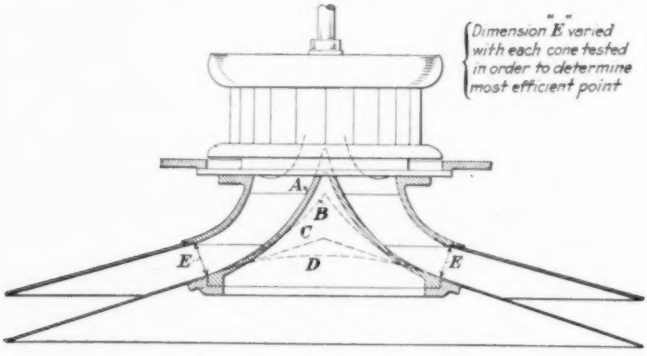


FIG. 8 HYDRAUCONE REGAINERS WITH CONE CENTERS AND CONICAL PLATES

21 Fig. 14 shows a series of efficiency curves plotted between horsepower and efficiency from test curves similar to those illustrated in Figs. 12 and 13. These curves give the efficiencies obtained at constant speed of the turbine throughout the range of the curves. The speed at 1 ft. head is indicated in the figure by the abbreviation r. p. $m._1$, the speed at best efficiency is different for each curve.

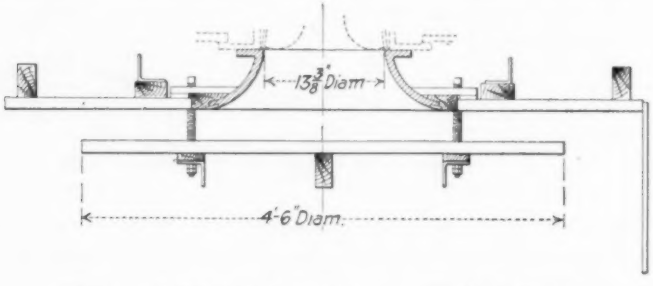


FIG. 9 HYDRAUCONE REGAINER WITH FLAT PLATE

22 It will be noted that the more efficient the regaining device, the higher the speed as well as the higher the power of the water wheel. It will further be noted that the increase of efficiency from "no draft tube" as shown in curve 1 to "hydraucone regainer" as

shown in curve 5 is from $77\frac{1}{2}$ per cent to 86 per cent, with an increase in speed from r. p. $m._1 = 155$ to r. p. $m._1 = 165$, and with a maximum power from $hp._1 = 0.270$ to $hp._1 = 0.310$. The great gain, therefore, by the use of hydraucone regainers is not only in higher efficiency but is also in both speed and power.

23 Attention is called to the fact that each of the curves shown in Figs. 14, 15, 16 and 17 is plotted from a complete test similar to those shown in Figs. 12 and 13, where a great many observations were plotted and smooth curves drawn.



FIG. 9A CONES SHOWN IN FIG. 8

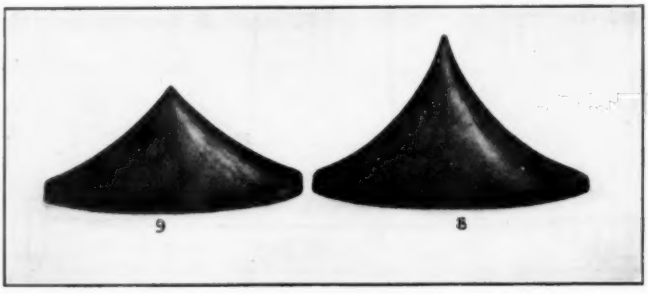


FIG. 9B CONES SHOWN IN FIG. 8

24 Curve 1, Fig. 15, shows the efficiency of the short curved draft tube, which, it will be noted, is less than with no draft tube at all. There is no question but what many plants in operation today are losing power because of the poor curved tubes they use.

25 Curve 5, Fig. 15, shows the efficiency of the power-house model of hydraucone regainer whose dimensions are given in Fig. 11 and are approximately the same as those of the hydraucone regainer installed at the Hiram plant of the Cumberland County Power & Light Company, Portland, Maine.

26 Curve 4 gives the results of a test of a plate-steel curved

draft tube that are very creditable, but it is very rare that there can be installed in a power house a draft tube of the same relative dimensions as those of the model employed, for the cost of excavation would be prohibitive.

27 Fig. 16 gives the results of tests with the hydraucone regainers shown in Fig. 8 and also with the straight tube and hydraucone regainer illustrated in Fig. 10. These curves are plotted from elaborate tests similar to those of Figs. 12 and 13 and show clearly that the hydraucone regainers with flat plates are slightly more efficient when tested with the model turbine than those with cone centers.

28 Fig. 17 presents the results of tests made on various shapes of conoidal chambers, from which it will be seen that the hydraucone

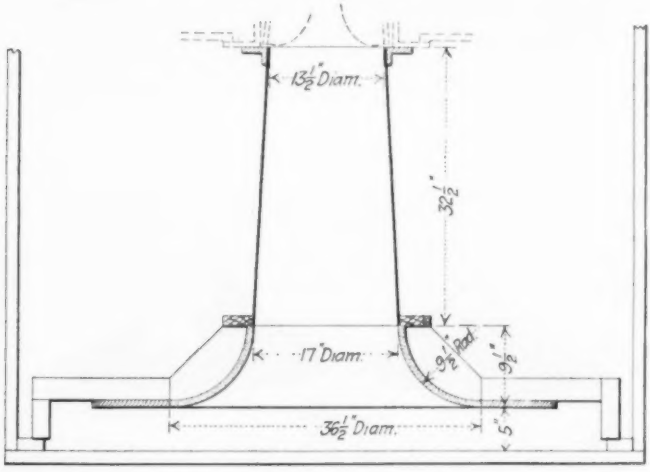


FIG. 10 COMBINATION OF STRAIGHT DRAFT TUBE AND HYDRAUCONE REGAINER WITH FLAT PLATE

regainer as here discussed is susceptible of a reasonable variation without materially affecting its efficiency. This series of tests bears out in a striking way the statement made at the beginning of the paper that the new device for regaining pressure from velocity consists of a conoidal chamber around the stream flow which gradually recedes from what would be the normal or free shape of the flow impinging upon the particular form of base used.

29 The early tests at the laboratory were made to determine the effects on the model water wheel of radial passages extending outward but with a slight angle downward from the horizontal,

and with passages leading directly from the runner. These early tests were not so successful as the later tests because it was not then realized that in order to change the direction of the water from axial as it flowed from the runner to radial and outward, it was necessary to take into consideration and provide for what was afterward called the "hydraucone action" of water, and which requires a definite curvature to properly change the direction of the water from axial to radical if the greatest regain of velocity head is to be expected and

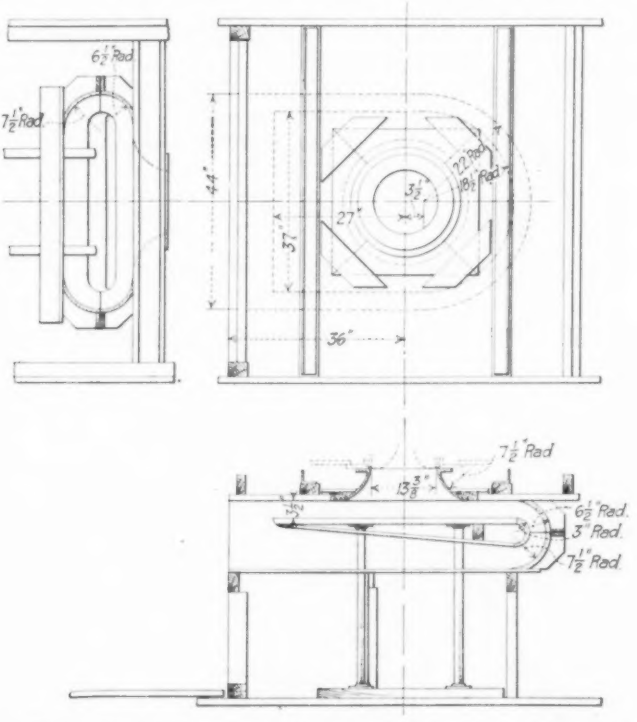


FIG. 11 DIMENSIONS OF MODEL OF HYDRAUCONE REGAINER USED IN A NUMBER OF POWER PLANTS

consequent losses are to be avoided. In searching for a means for increasing the efficiency of regain, the author recalled some experimental work done in connection with the determination of the pitot-tube formula in 1900, the results of which were published in Vol. 22 of the *Journal* of the Association of Engineering Societies (1901), page 284. Fig. 21 of that publication shows the

stream lines and velocities of a jet impinging upon a plate, and by reason of the importance of the principle there disclosed in the new

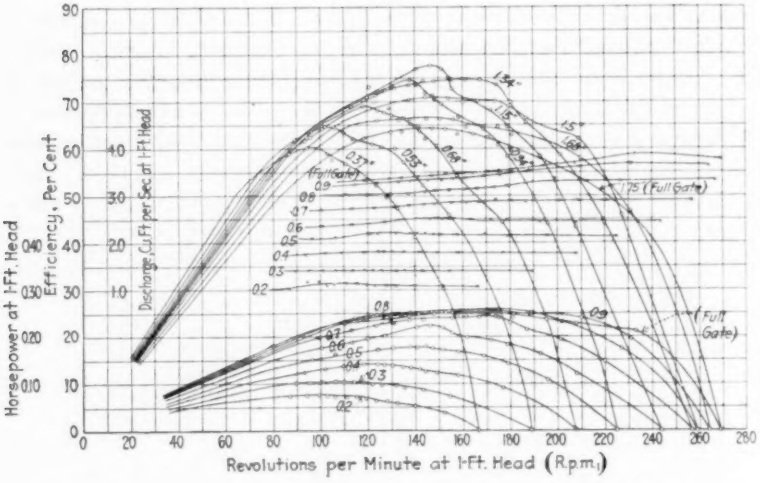


FIG. 12 CURVES OF TEST OF MODEL OF ARGO CURVED DRAFT TUBE NO. 1

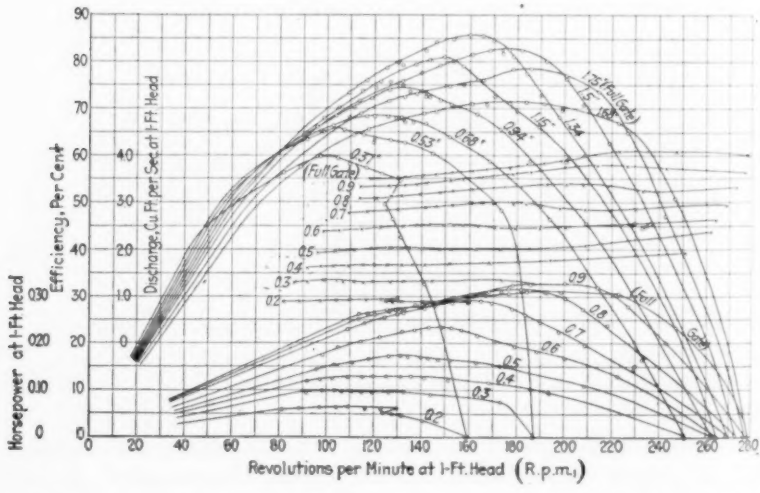


FIG. 13 CURVES OF TEST OF MODEL OF HYDRAUCONE REGAINER WITH FLAT PLATE

(7½-in.-radius conoidal chamber 4 ft. 6 in. in diameter.)

method of regaining pressure from velocity, it is reproduced here as Fig. 18.

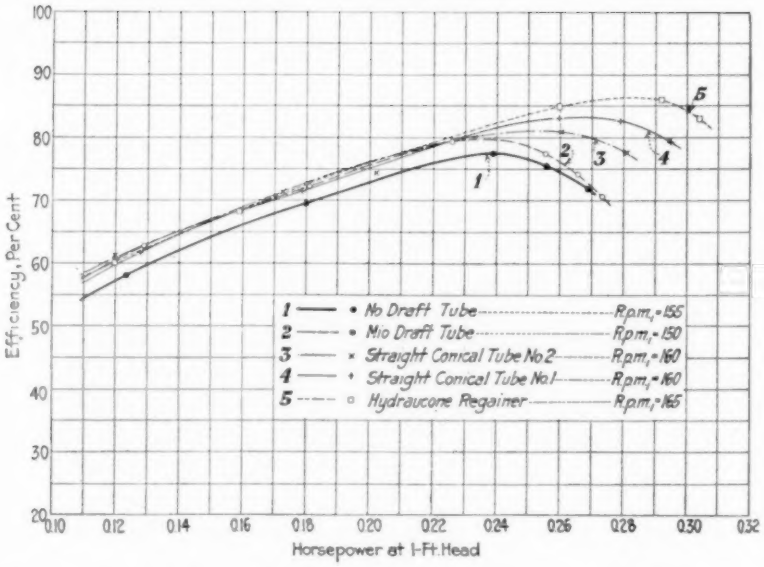


FIG. 14 CURVES SHOWING RELATION BETWEEN EFFICIENCY AND HORSEPOWER AT 1 FT. HEAD

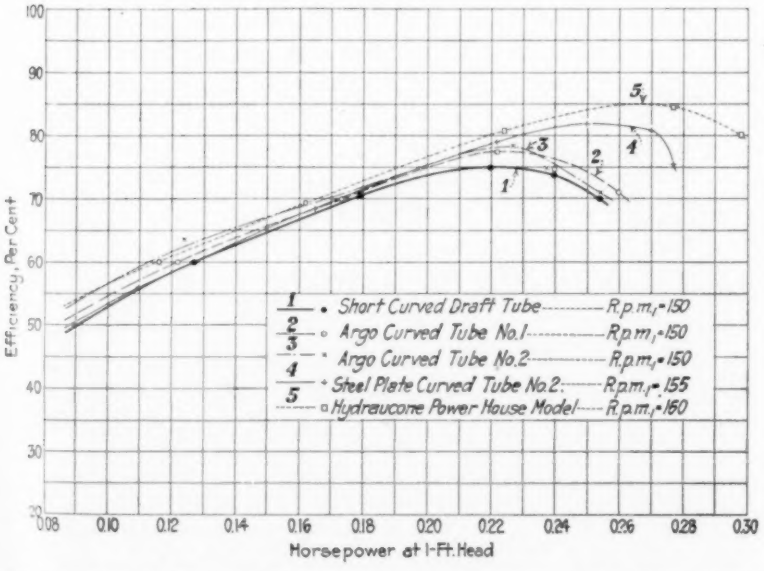


FIG. 15 CURVES SHOWING RELATION BETWEEN EFFICIENCY AND HORSEPOWER AT 1 FT. HEAD

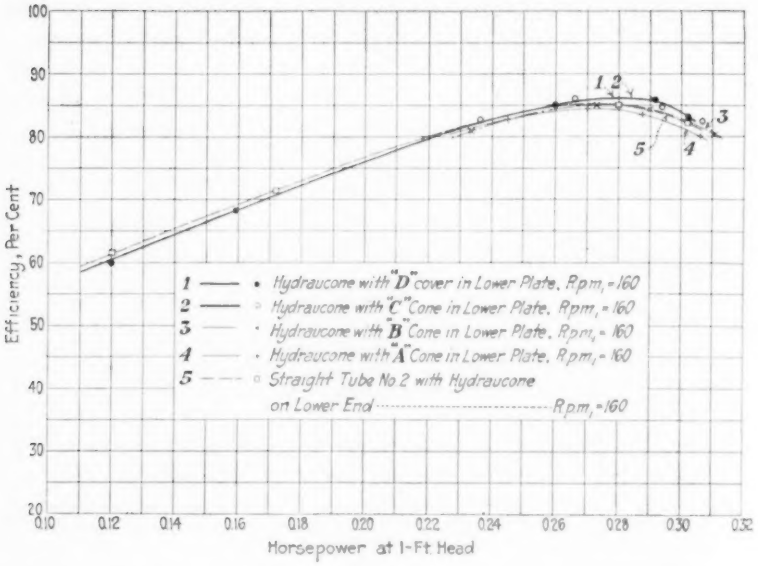


FIG. 16 CURVES SHOWING RELATION BETWEEN EFFICIENCY AND HORSEPOWER AT 1 FT. HEAD

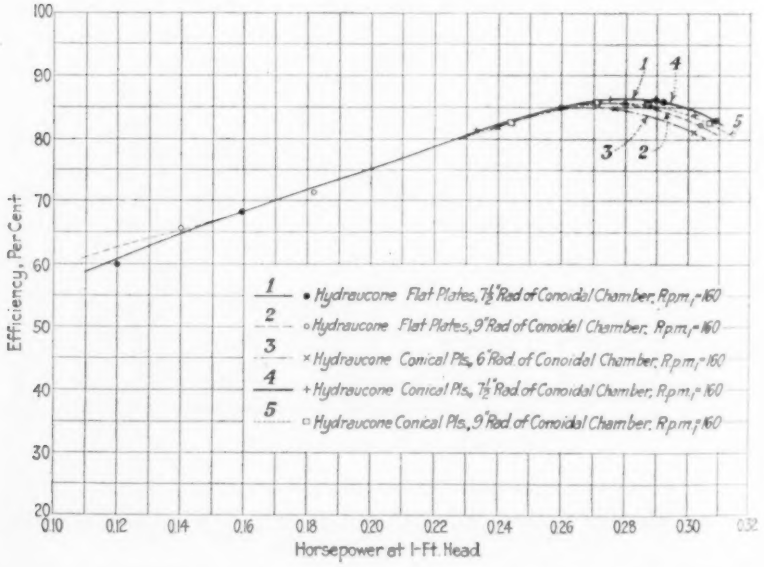


FIG. 17 CURVES SHOWING RELATION BETWEEN EFFICIENCY AND HORSEPOWER AT 1 FT. HEAD

30 The lines in this figure represent the stream lines of flow of the water, which were traced by injecting a small stream of blue water into the jet at various points along a diameter and noting and recording the flow line. The plate used in this case was of glass so as to afford ample light for making the observations. The figures in the circles are the velocities in feet per second at the various locations indicated. It will be noted that the water at the center of the jet slows up as it reaches the plate and delivers its velocity head into pressure head at the center of the plate. This pressure head at the center is transformed into velocity as this stream flow passes from the center of the point of impingement to the point beyond the radius

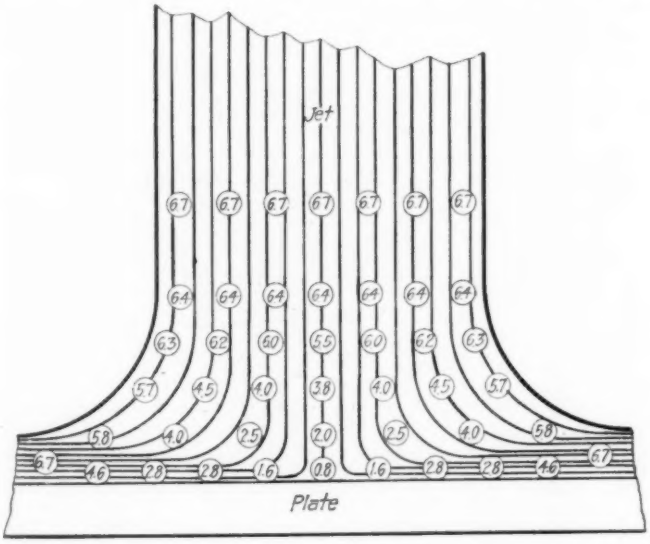


FIG. 18 HYDRAUCONE ACTION OF WATER WHEN A JET IMPINGES NORMALLY ON A FLAT PLATE

of curvature. Referring to Fig. 18, the radius of curvature of the water in making its change in direction is a definite curve and is the minimum which can be used in regaining devices and yet obtain the reactions within the point of impingement which will give smooth stream flow of discharge along the plate and consequently without eddies and attendant losses.

31 By the term "hydraucone action" of water is meant that action which occurs when an unenclosed stream impinges against and is deflected along a surface. Fig. 18 illustrates the hydraucone

action of water when a jet impinges normally upon a flat plate. In considering the three outer stream filaments it was noted and recorded in 1900 that the cross-sectional area of the flow increased in making the turn and decreased at the exit from the turn. By placing a cone on the plate with its axis in line with the center line of the jet, the outer shape of the curvature of the water is changed, but not to a great extent because of the low velocities within the center portion of the impinging jet.

32 When the hydraucone action of the water as illustrated in Fig. 18 was taken into consideration in the laboratory work, conoidal

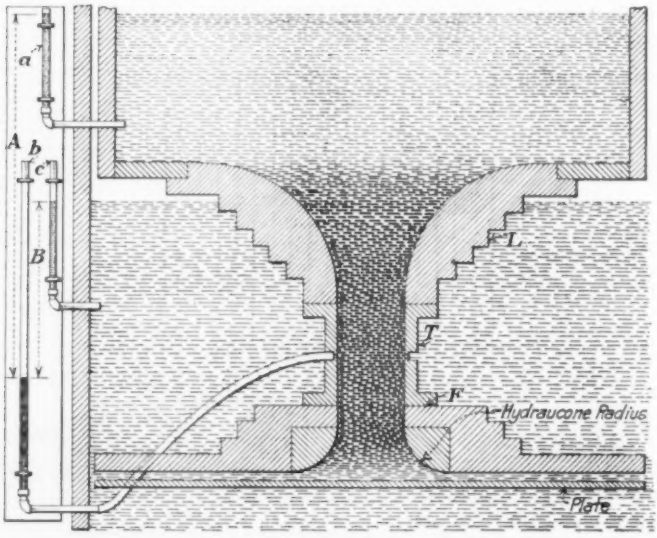


FIG. 19 APPARATUS USED IN TESTING SMALL MODEL DRAFT TUBES AND HYDRAUCONE REGAINERS

chambers were designed such as the one shown in Fig. 9, and when these were combined with the radial passages the desired results were immediately secured.

TESTS ON SMALLER MODELS

33 The tests of the various model draft tubes and hydraucone regainers made at the laboratory with the small turbine illustrate the practical, economical, and efficient application of the hydraucone

principle for increasing the efficiency of water-power developments. Coincident with the series of tests in connection with the small turbine, a series of tests on smaller models were made without the use of a model water wheel. Sometimes these small models were tested before the large models were built and tested with the turbine, and sometimes afterward. These models were all made of such size that the entrance was 4 in. in diameter at the throat as compared with $13\frac{3}{8}$ in. at the throat of the larger models.

34 The apparatus used for testing the small models is shown in Fig. 19. The model water wheel was removed from the plate-steel cylindrical casing shown in Fig. 2 and in its place the curved wooden section L was bolted to the opening at the bottom of the casing. At the bottom of the curved piece L was connected a 4-in. diameter

TABLE 1. EFFICIENCIES OF VARIOUS SIZES OF STRAIGHT CONICAL MODEL DRAFT TUBES

Tube No.	Small Diameter, in.	Large Diameter, in.	Length, in.	Efficiency, per cent
1	4	8	$9\frac{1}{2}$	41
2	4	6	$27\frac{1}{2}$	65
3	4	8	$27\frac{1}{2}$	76
4	4	8	$21\frac{1}{2}$	61
5	4	$5\frac{1}{2}$	9	46
6	4	6	9	60

throat T, made of brass pipe with flanges, to the lower end of which the various small models were fastened. Four piezometer openings and as many gages were provided to indicate accurately the pressure at the throat. Gages a, b, and c serve to illustrate more or less diagramatically the method of computing the efficiencies of the small models. The difference in elevation between gages a and c was kept about the same. The reading of the throat gage b varied widely, depending upon the efficiency of the regainer device used. The efficiency of these small models was taken as B divided by A, or in other words, the velocity head regained divided by total velocity head in the water as it passed through the throat T. A small model hydraucone with large-diameter plate is shown attached to the throat in Fig. 19. The model hydraucone was detachable from the flange F of the throat, to which were bolted the various devices tested.

35 Straight conical model draft tubes of various sizes and proportions were tested, and their efficiencies computed by the method set forth above are given in Table 1. Cone No. 5 is a model of the straight conical draft tube No. 2 used with the model water wheel, and No. 6 is a smaller model of straight conical draft tube No. 1.

36 A 4-in. model of the short curved draft tube of Fig. 5 tested with the water wheel was made and tested, and its efficiency was found to be zero. A small model of Argo draft tube No. 1 was made as illustrated in Fig. 20. Its diameter at entrance was 4 in. and its various other dimensions can be computed by reference to Fig. 6. This model draft tube was attached to the flange F of the apparatus in Fig. 19 and a test made which showed an efficiency of 4.63 per cent. The model draft tube in the test was not far below the surface of the water, so that the hand could be inserted in the discharge end of

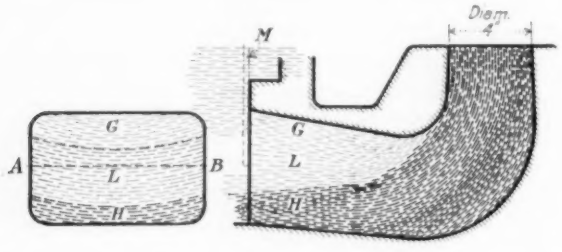


FIG. 20 FLOW CONDITIONS IN ARGO DRAFT TUBE NO. 1

the tube and the velocities traced by the sense of touch. Pitot tubes were also used to determine the velocities at various points. The heavily shaded portion shown in Fig. 20 represents high-velocity water. There was no flow at L, but the water flowed upstream or back in the tube in the region G as shown. The flow conditions in this draft tube illustrate very strikingly the fact that when the walls recede too rapidly from the stream flow the latter breaks away from the inner wall and flows at high velocity on the opposite wall.

37 A testing device as illustrated in Fig. 19 provided smooth stream flow from the throat T, so that the flow conditions were symmetrical about the axis of the models tested. When, however, the larger models were tested with the small water wheel, the water was discharged from one side or the other of the draft-tube outlet, depending upon the direction of the whirl of the water as it left the runner. At low speeds and high gate openings the water usually

whirls in a direction counter to the rotation of the runner, and at small gate openings and high speeds, with the rotation of the runner. The whirling action of the water in a curved tube as here illustrated tends to increase the efficiency of the tube.

38 It will be noted that the upper surface of the region *H* of high velocity is curved, as shown at the left in Fig. 20. This is caused by velocity of the water being greater at the center of the jet issuing from the throat *T* than at the sides, and consequently the surface is depressed as indicated by reason of the greater centrifugal force of the center portion of the water due to its higher velocity as it makes the turn.

39 An interesting experiment was performed by placing a board *M* over the end of the draft tube and bringing it down to the

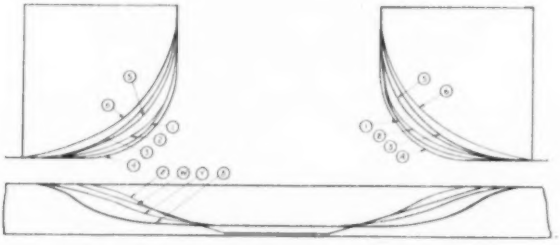


FIG. 20A SOME OF THE TYPES OF REGAINERS TESTED

(1) and (2), shaped to templet; (3), 2¾ in. radius; (4), 1¾ in. radius; (5) and (6), free hand

line *A-B*, when it was found that the efficiency of the tube was increased to 11.6 per cent. The same experiment was made by placing boards in the same relative position on the 13⅜-in. model draft tube used with the turbine, and while some increase in efficiency was obtained, it was not in the ratio here shown.

40 Similar flow conditions were found in the 4-in. model of the Argo draft tube No. 2. The efficiency of this draft tube was found to be 4.52 per cent, and with a board in the position shown in Fig. 20, it was increased to 12.45 per cent.

41 The 4-in. model of the plate-steel curved draft tube mentioned in Par. 26 when tested under the conditions set forth above showed an efficiency of 41.8 per cent. By placing a board over a portion of the discharge end in the same relative position as shown in Fig. 20, the efficiency was increased to 43.7 per cent.

42 Tests were made with the hydraucone regainers having cones in the center of the chamber, and the distance between the plates was varied. With a tall cone in the center similar to *A*, Fig. 8, the efficiency obtained was less than 70 per cent. By substituting a low cone similar to that represented by *B* in the same figure, the maximum efficiency by varying the distance between the plates was found to be 70 per cent.

43 One series of tests copied from the record book of a model hydraucone regainer as shown in Fig. 19 having a $2\frac{3}{4}$-in. radius of curvature of the conoidal chamber, gave the following efficiencies as the distance between the plates was varied:

Distance between plates, in.	1/2	17/32	9/16	19/32	5/8	3/4	7/8
Efficiency, per cent.	76.7	77	78.5	78.7	78	74	68

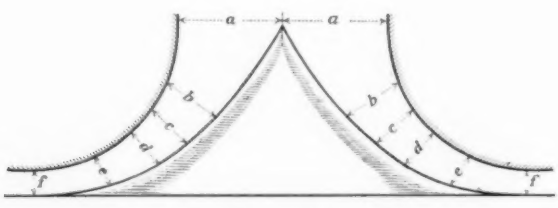


FIG. 21 EQUAL-AREA CONE

By cutting down the diameter of the upper and lower plates to 18 in., using the same radius of curvature of the conoidal chamber, namely, $2\frac{3}{4}$ in., and varying the distance between the plates, the following efficiencies were obtained:

Distance between plates, in.	1/2	9/16	5/8	11/16	3/4	7/8
Efficiency, per cent.	61	65.6	67.5	67	68.8	59.3

44 Conoidal chambers of various radii of curvature were tested with cone centers, flat plates, and then with depressions or concave surfaces, the latter being developed to secure shorter radius of curvature of the conoidal chamber. (See Fig. 20*a*.) The efficiencies obtained with the concave surfaces were about equal to that obtained by the use of the flat plate when the proper curve of the conoidal chamber was used. The results obtained in a test using curves (3) and (X) of Fig. 20*a* are as follows:

Distance between plates, in.	1/2	9/16	5/8	11/16	3/4	7/8	1 1/8
Efficiency, per cent.	60.2	73.3	74	73.5	71	67.7	66

45 Fig. 21 shows what was termed an "equal-area cone." The conical sections *a*, *b*, *c*, *d*, *e* and *f* were of equal area when the dimension *f* was made $\frac{1}{2}$ in. A test of the efficiency of this device was made on December 16, 1914, with the apparatus illustrated in Fig. 19, and the following efficiencies were obtained, using a conoidal chamber of $2\frac{3}{4}$ in. radius:

Distance between plates, in.	½	9/16	5/8	11/16	¾	7/8
Efficiency, per cent	71.3	72.4	72.8	72.4	70	64.7

46 Fig. 22 shows the flow conditions in straight conical regainer No. 6, Table 1, when water was discharged into its small end from a 4 in. elbow having a 4 in. radius of curvature, and when connected to the testing device, Fig. 19. The efficiency of the tube under this

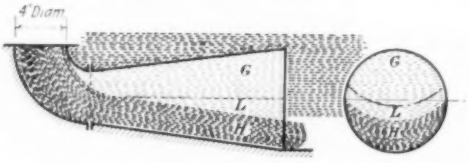


Fig. 22 Flow Conditions in Straight Conical Regainer No. 6

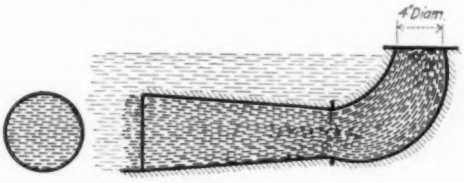


Fig. 23 Equalizer Bend and Straight Conical Draft Tube

condition was found to be zero. When an elbow with an 8-in. radius was substituted for the 4-in. radius elbow, the efficiency rose to 28 per cent. Considerable experimenting was done to find an elbow which would give flow conditions at the small end of this straight conical tube such that high efficiency of regain could be obtained. These experiments resulted in the development of the elbow illustrated in Fig. 23 which was called the "equalizer bend." The area of the elbow is greater at its mid-sections than at either end. The outer filaments of flow in Fig. 18 gave indications of the natural tendency of the water to require larger area at the mid-section of a bend than at either end, and this fact was taken into consideration in all designs and experiments.

47 When a combination consisting of the equalizer bend and tube No. 6 (Table 1) as illustrated in Fig. 23 was tested with the

testing apparatus shown in Fig. 19, its efficiency was found to be 60 per cent. The mean radius of curvature of the elbow was 7 in. By referring to Table 1 it will be seen that the efficiency of tube No. 6 without the elbow was 60 per cent.

48 A larger model of the equalizer bend was made for testing with the small water wheel. With one end of the equalizer bend connected to the water wheel and the other end connected to the straight conical draft tube No. 1, Fig. 4, the efficiency of the water wheel with this combination was found to be 81 per cent. With the equalizer bend removed and the draft tube connected directly to

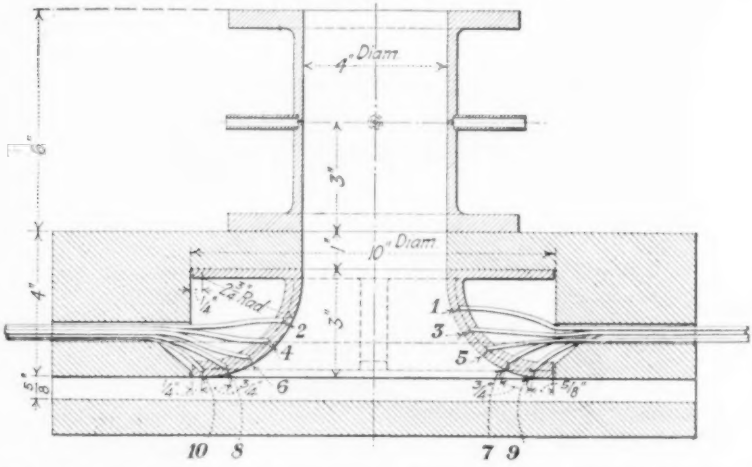


FIG. 24 ARRANGEMENT OF PIEZOMETER CONNECTIONS IN TESTS TO DETERMINE PERCENTAGE OF REGAIN ALONG CURVED SURFACE OF CONOIDAL CHAMBER

the water wheel, the efficiency was found to be 82.7 per cent. The equalizer bend affords a means of turning the water at right angles with better conditions for regain in a straight-axis tube than any of the other draft tubes which were developed.

49 Tests were made to determine the percentage of regain along the surface of curvature of the conoidal chamber, piezometers being attached as indicated in Fig. 24. These tests showed conclusively that there is a regain of pressure from velocity as the water makes the turn from an axial to a radial direction.

50 Fig. 25 indicates the relative regain for each $\frac{3}{8}$ in. measured along the curvature of the conoidal chamber walls. It was found

that the regain was substantially the same for each unit of linear dimension when the proper shape of conoidal chamber was used for the particular base upon which the flow impinged.

51 The foregoing discussion has been confined to the simple shapes tested. Tests were also made upon square expanding nozzles, and elliptical-shaped nozzles as contrasted to the circular straight nozzles. Rectangular and conoidal bends were also tried out.

HOLYOKE TESTS

52 At the completion of the tests of the various regainers and model draft tubes at the laboratory, a series of tests were made at the Holyoke Testing Flume on a large water wheel having a runner of

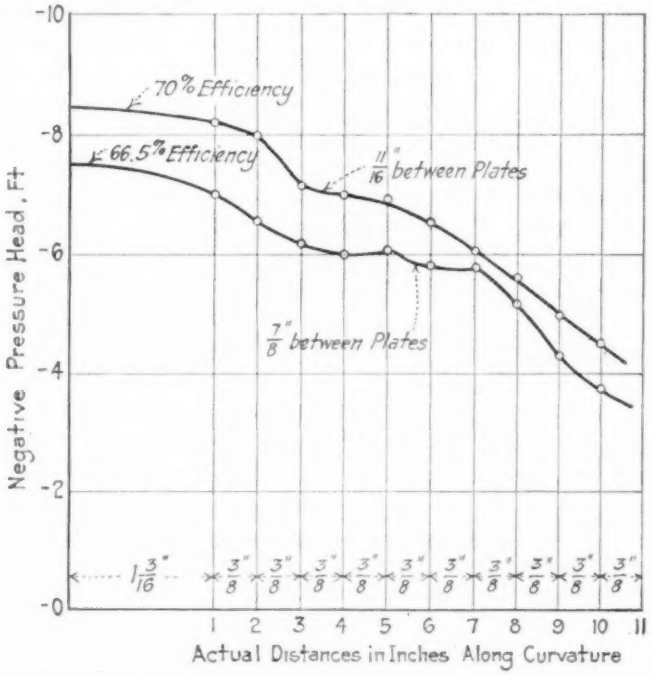


FIG. 25 RELATIVE REGAIN PER UNIT OF LENGTH ALONG CURVED SURFACE OF CONOIDAL CHAMBER

30 in. nominal diameter and a discharge diameter of 38 in. The objects of these tests were to verify the results already obtained at the laboratory, and to obtain records of the performance of the

hydraucone regainer which could be presented to the engineering profession with the assurance that they were reliable and could be used for comparison with the results obtained with other forms of draft tubes tested at Holyoke. The water wheel was first tested with a straight draft tube 8 ft. $3\frac{1}{8}$ in. in length and with diameters of 3 ft.

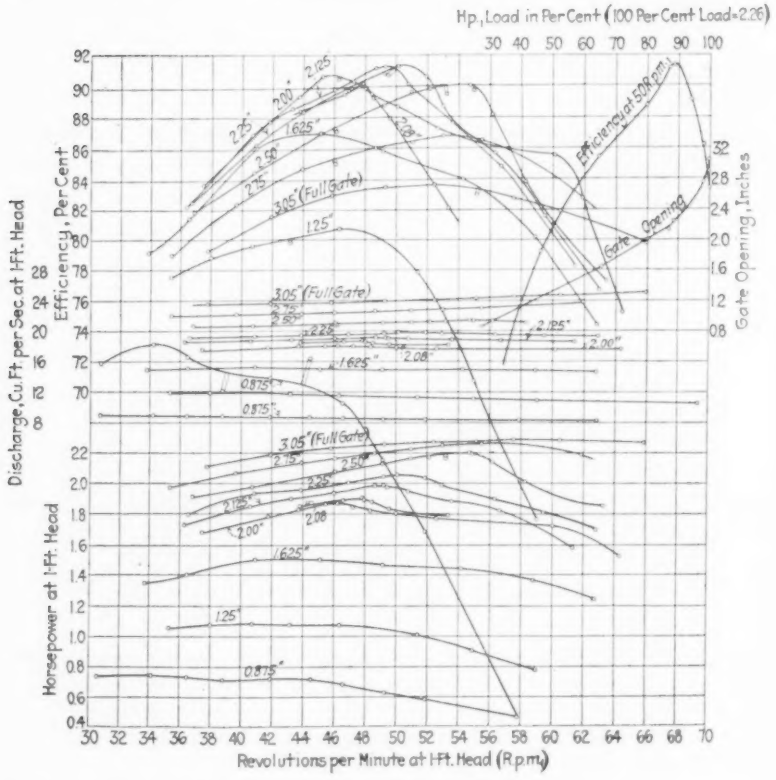


FIG. 26 HOLYOKE TEST NO. 2445 OF 30-IN. RUNNER WITH LONG STRAIGHT CONICAL DRAFT TUBE

(20 guide vanes; guide case, $14\frac{1}{4}$ in. wide)

$3\frac{1}{2}$ in. and 6 ft. 3 in. at the smaller and larger ends, respectively. This was the best tube of this type developed by the Allis-Chalmers Mfg. Co. for testing purposes. The results of the test are shown in Fig. 26, Holyoke test No. 2445, dated March 3, 1916.

53 The next test was made by placing the water wheel directly

on the hydraucone regainer as shown in Fig. 27, and the results are given in Fig. 28, Holyoke test No. 2460, dated May 5, 1916. Fig. 29 gives the results of a test of the water wheel when set on a short straight conical draft tube (3 ft. long), with the discharge of the tube connected with the hydraucone regainer. (Holyoke test No. 2461, dated May 16, 1916.)

54 Two shapes of conoidal chambers were made and tested, one of which is illustrated in Fig. 27. Tests were also made with the plates of the hydraucone regainer set at various distances apart. These tests confirmed those made at the laboratory and showed that the hydraucone regainer could be relied upon to give high efficiency with commercial water wheels under actual operating conditions. A maximum efficiency of 91.6 per cent was obtained with the use of the hydraucone regainer as shown by Holyoke test No. 2461, which

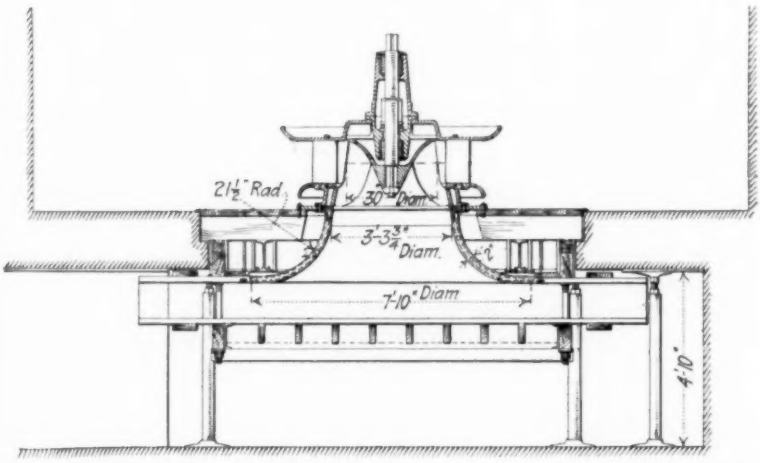


FIG. 27 WATER WHEEL PLACED DIRECTLY ON HYDRAUCONE REGAINER AS IN HOLYOKE TEST NO. 2460

was slightly greater than that obtained with the long straight draft tube in Holyoke test No. 2445. The test of the water wheel with the hydraucone regainer showed better characteristics at speeds both below and above normal than the test with the straight draft tubes, all of which might be expected in view of the radial passages of the hydraucone regainer in which the whirling action of the water is utilized for increasing the efficiency.

55 Hydraucone regainers with cone centers such as *A*, Fig. 8, were not tested at Holyoke because the laboratory experiments showed conclusively that hydraucone regainers with flat plates gave better efficiency. The friction of the water along the surface of a hard cone center is much greater than the internal friction when the water forms its own cone as shown in Fig. 18. This is one reason why the hydraucone regainer with flat plate gives better efficiency than the hydraucone regainer with cone center.

COMPARISON OF RESULTS TO BE OBTAINED BY THE USE OF THE HYDRAUCONE REGAINER WITH RUNNERS OF DIFFERENT SPECIFIC SPEEDS

56 The hydraucone regainer may be employed in any power plant operating under any head, with a resulting increase in efficiency. Its actual value in power plants operating under different heads with runners of different specific speeds may be seen from Table 2.

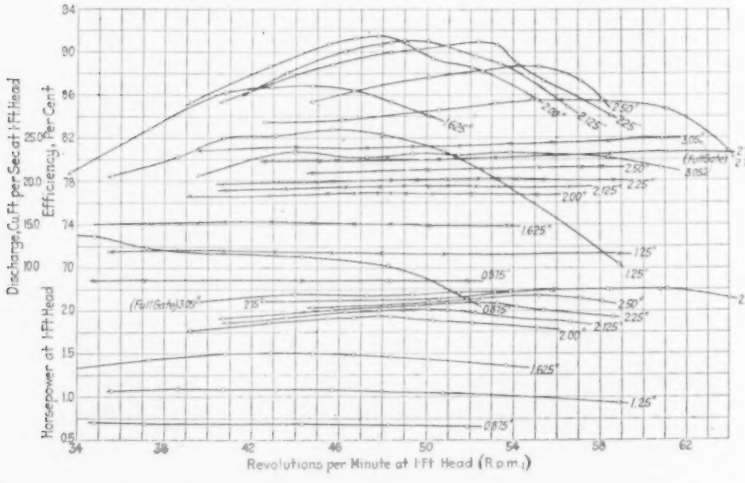


FIG. 28 HOLYOKE TEST NO. 2460 OF WATER WHEEL PLACED DIRECTLY ON HYDRAUCONE REGAINER

(Radius of conoidal chamber, 21½ in.; distance between plates, 12½ in.)

57 Runner No. 1 is a type identical with that used on one of the 37,500-hp. turbines recently put into operation in Station No. 3 Extension of the Niagara Falls Power Company. The specific speed of this runner is about 40 (English system). Runner No. 2 is a reasonably high-speed runner for low-head plants, such as that

installed at the Argo plant of the Eastern Michigan Power Company, and it is the same type as that used in the experiments made at the laboratory. Runner No. 3 is the Allis-Chalmers No. 404 high-speed type furnished to the Cheboygan Electric Light and Power Company, Cheboygan, Mich. This runner was described by Mr. Nagler in his

TABLE 2 INCREASE IN POWER-PLANT EFFICIENCY RESULTING FROM USE OF THE HYDRAUCONE REGAINER

1	2	3	4	5	6	7	8	9	10	11	12
No of runner	Specific speed	Nominal diameter of the runner, in.	Discharge diameter of the runner, in.	Cu. ft. per sec. discharged at 1 ft. head	Vertical component of water discharged from the runner	Energy in discharged water, per cent.	Assumed efficiency of draft tube, per cent.	Loss of energy with draft tube, per cent.	Assumed efficiency of hydraucone regainer, per cent.	Loss of energy by hydraucone, per cent.	Increase of efficiency by use of hydraucone, per cent.
1	40	30	28	1.97	1.65	4	40	2.40	70	1.2	1.2
2	90	30	44.5	32.05	2.97	13.7	20	11	70	4.1	6.9
3	160	30	30	18.5	3.8	22.4	20	17.9	70	6.7	11.2

paper presented at the 1919 Annual Meeting of the Society.[1] The axial component of the velocity of the water discharging from each runner is shown in column 6. The energy discharged from the runner in each case is shown to be respectively 4 per cent, 13.7 per cent, and 22.4 per cent of the energy of the total head. Draft tubes on high-head plants are usually much more efficient than those on low-head plants. A draft-tube efficiency of 40 per cent may be obtained under ideal conditions in a suitable curved draft tube for use in connection with runner No. 1, but draft tubes on low-head plants are not so efficient, and the efficiency of a curved draft tube such as would be used with runner No. 2 would be about 20 per cent, and with runner No. 3 certainly not exceeding 20 per cent.

58 The efficiency of the hydraucone regainer for use with each of the three runners would be at least 70 per cent. The greater efficiency of the hydraucone regainer over the curved draft tube results in increasing the power-plant efficiency by 1.2 per cent, 6.9 per cent, and 11.2 per cent, respectively, for runners Nos. 1, 2, and 3,

[1] A New Type of Hydraulic-Turbine Runner, Forrest Nagler, Trans. Am. Soc. M. E., vol. 41, p. 829.

as shown by column 12 of Table 2. From this it will be seen that the use of the hydraucone regainer on low-head plants is almost a neces-

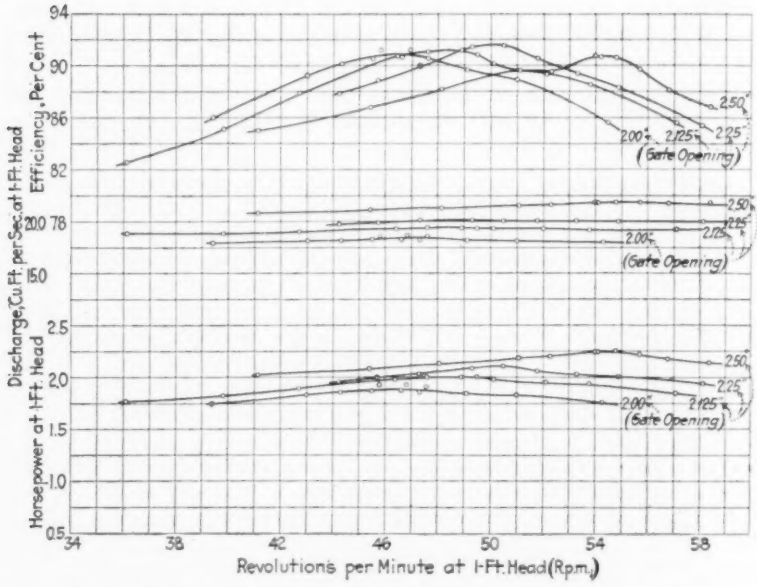


FIG. 29 HOLYOKE TEST NO. 2461 OF WATER WHEEL SET ON SHORT STRAIGHT CONICAL DRAFT TUBE AND DISCHARGING THROUGH A HYDRAUCONE REGAINER

(Draft tube of 3 ft. section; radius of conoidal chamber, 25 in.; distance between plates, 12½ in.)

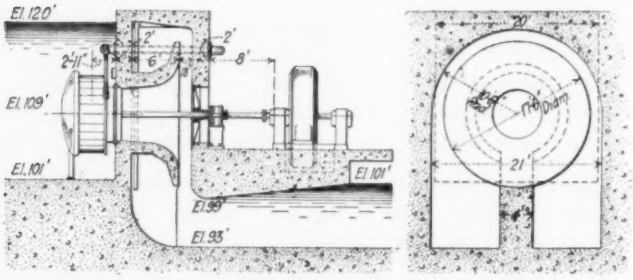


FIG. 30 HYDRAUCONE REGAINER USED WITH WATER WHEEL HAVING A HORIZONTAL SHAFT

sity if reasonably good efficiencies are to be obtained. These comparative efficiencies are based on the wheels operating at their best speed and best efficiency, that is, when the discharge from the runner is nearly axial. At any other speed or gate opening the discharge of

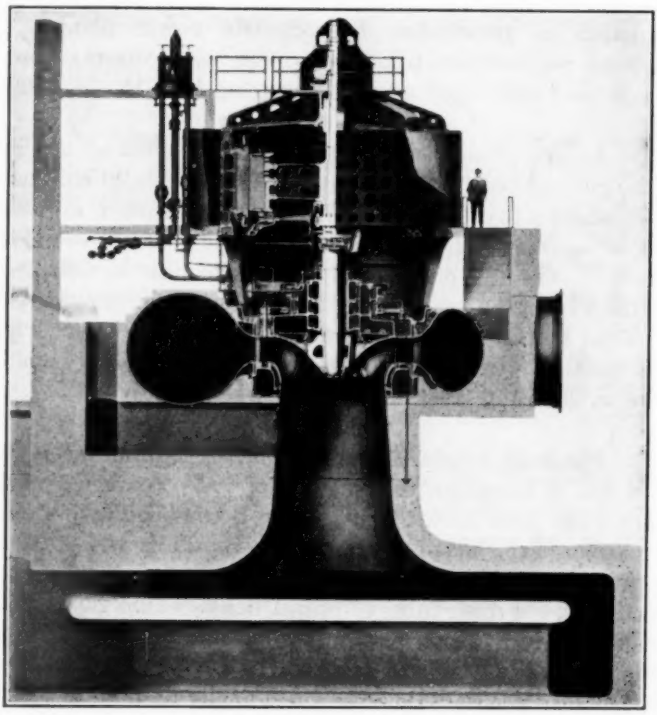


FIG. 31 HYDRAUCONE REGAINER OF FLAT-PLATE TYPE INSTALLED IN STATION NO. 3 EXTENSION OF THE NIAGARA FALLS POWER COMPANY

the water wheel is accompanied by more or less whirling, that is, the absolute velocity of the water leaving the runner is much greater at full load than that given in Table 2. Furthermore the whirling action of the water in the discharge is much more effectively used in the hydraucone regainer than it is in the curved draft tube because of the radially extending passages within which this whirling action produces outward pressure by centrifugal force.

FIELD OF APPLICATION OF THE HYDRAUCONE REGAINER

59 The field of application of the hydraucone regainer will be greater on low-head plants, although it is now being installed in connection with two 40,000-hp. units which are to operate under a head of 421 ft. Hydraucone regainers are in operation or in course of construction in twenty-five hydroelectric power plants. These hydraucone regainers are to operate under heads varying from 8 ft. to 421 ft. and with water wheels developing from 150 to 40,000 hp.

60 An interesting installation is that at the plant of the Cheboygan Electric Light and Power Company, Cheboygan, Mich., shown in Fig. 30, where a hydraucone regainer was used with a water wheel having a horizontal shaft. The hydraucone passages were placed vertically. The top of the hydraucone chamber is nearly 19 ft. above the level of the water in the tail race, but no difficulty has been experienced in the effective operation of the hydraucone regainer in this position. Others of this type have either been installed or are now in course of construction.

61 The most notable installation of the hydraucone is that in Station No. 3 Extension of the Niagara Falls Power Company at Niagara Falls, New York. Unit No. 16 is equipped with a hydraucone regainer of the flat-plate type, as shown in Fig. 31. At the time of the purchase of the water wheel for this unit it was still argued by some that curved draft tubes of proper design would give as high an efficiency as that claimed for the hydraucone regainer. The engineers of the Power Company accordingly conducted a long series of experiments with various devices which finally resulted in the development of a conoidal chamber with a cone center similar to that shown in Fig. 8 for units Nos. 17 and 18. The best model with cone center had the proper-shaped conoidal chamber for regain and when the cone center was removed from the model it was found that the efficiency was no lower than with the cone in place. The use of the new substitutes for curved draft tubes at this plant resulted in a water-wheel efficiency of 93 per cent being obtained, which is corroborative evidence of the value of the hydraucone principle for the regaining of pressure from velocity.

62 Every one familiar with hydroelectric power-house operation where curved draft tubes are employed has frequently noticed the

boiling and disturbance of the water at the outlet from the curved tubes. This disturbance is clearly an evidence of inefficiency and loss of energy which should be more effectively utilized. The hydraucone regainer provides a means for efficiently regaining the energy discharged from the runner for useful effect upon the water wheel within the limited space available in the power-house foundations.

DISCUSSION

GEO. R. SHEPARD. In the fall of 1917, the then Hydraulic Power Company was required by the government to proceed with the rapid development of additional power at Niagara Falls for use for war purposes. The engineers of the company, having foreseen this requirement, were actively engaged in the prepartion of plans for further development. Preliminary specifications and proposals had been asked from the two leading American water-wheel companies. During the preliminary discussions of the type and size of unit to be adopted, the author recommended and urged the adoption of the hydraucone regainer, described in his paper, for the purpose of increasing the efficiency of the water wheels.

The device was so new and novel, and the claims for it were so vigorously put forth, that it did not seem to the Power Company engineers that the opportunity should be passed by of securing better efficiency if it could be obtained.

The competing company presented designs embodying their curved form of draft tube which had been carefully worked out, through years of experience, and, since the engineers of the competing company were positive that the efficiency of their curved draft tube was so high that there could be no material gain in the adoption of the hydraucone, the engineers of the Power Company determined, at the suggestion of the author, to conduct a series of experiments to settle in their minds, whether the hydraucone was sufficiently superior in efficiency to the best forms of curved draft tubes to justify its adoption in the new station.

Accordingly, the Allis-Chalmers Manufacturing Co. presented to the Power Company engineers a model hydraucone similar to the one finally adopted, and also the best form of their curved draft tube. The competing company also supplied a model of their best curved draft tube, made to the same scale as the hydraucone and the other tube.

The Hydraulic Power Company constructed a temporary laboratory with pumps, tanks and measuring weirs, necessary to secure a comparison of the models presented. The testing equipment was substantially the same as that shown in Fig. 19 of the author's paper. The early tests showed such superiority of the hydraucone model, that the engineers of the Power Company were convinced of the necessity of adopting some such form for the new station.

It should be borne in mind that the space underneath the turbine was limited within which an acceptable form of tube had to be devised. Various models were worked out by the engineers of the competing company, were tested, and the Power Company's engineers also designed and tested some models of their own.

The records of this series of tests are interesting but too long to recite fully here. It will probably be of interest to mention some of the forms tried out.

a One form was the straight draft tube — as long as could possibly be obtained within the space available.

b Another form was similar to this straight tube, except flared partly at the end and fitted with a cone in the center.

c Another form was that of a curved draft tube, very much flattened along the horizontal portion, in an attempt to conform to the natural shape of the flow lines of the water, which it assumed in going around the bend. The flow lines in the curved draft tube were found to be substantially as shown in Fig. 20 of the paper under discussion and the flattened curved tube was made for the purpose of conforming, as nearly as possible, to the flow shape, in an attempt to secure results comparable to the hydraucone.

The final alternate design adopted, known as Model "M," was designed by Mr. Lewis F. Moody, Consulting Engineer of the Wm. Cramps Ship and Engine Building Company, and was of a symmetrically flared radial-discharge type, having a cone in the center, somewhat similar to Fig. 8 in the paper. The hydraucone type was used on the Allis-Chalmers unit and the model "M" type on the other units.

Model "M" was found to be greatly superior to either of the types *a*, *b* and *c* noted above.

It should be borne in mind that the small model was of 4 in. diameter intake, similar to that of the small models discussed in Par. 33 of the paper.

The tests of the small models led the Power Company engineers to believe that the Model "M" with the cone in the center, while

it did not give quite as high an efficiency as the model without the cone in the center, seemed to offer more stability than the other — that is to say, the models giving the highest efficiency did so only at a critical velocity, the efficiency falling off rapidly at greater or less discharges. These more efficient models were also very sensitive in regard to the dimensions marked " 5 " in the author's Fig. 10 — a slight variation in this dimension making a difference in the regainer efficiency. The small model with flat plate was found to be more sensitive in this respect than the one having a conical center.

The writer has noted with interest, however, the records of tests in the paper of the symmetrically flared type with a model water wheel, with and without cone center, and apparently the author has not found the sensitiveness under actual working conditions which was found in the small models. It is probable that further practical experience will show that this sensitiveness was due to the small models used and will not be found in actual practice.

The final alternate model adopted was tested after removing the cone center and it was found that the efficiency of this model under that condition was no less than with the cone center in place.

It should be understood that these model tests were made under ideal conditions, with straight flow lines, as indicated in Fig. 19 of the paper.

The company engineers did not have the advantage of determining the effect of whirling action upon the models or of any other disturbances caused by an actual water-wheel runner, and therefore their tests did not prove that the relative results secured would hold true in actual practice.

In addition to the gain in efficiency there is a practical advantage in the use of the radial-discharge type of tube due to the saving of excavation. This excavation is below the tailrace water elevation and in most instances, the company's particularly, the problem of unwatering is difficult and expensive. The width of tailrace required is fixed by the diameter of the bottom plate necessary to reduce the discharge velocity to the required amount. In this case the residual discharge velocity head was about 0.25 feet.

The design of the forms for the construction of the draft tubes required some careful consideration in the drafting room and the construction force required some education; however, the first one was constructed without too much difficulty and also without accident. The second one was made in about the same time as the first, while the third was made in a much shorter time and with the same

facility as the ordinary form work of the station. The first one to be constructed was as shown in Fig. 31, and the second and third somewhat similar to Fig. 8. It was impossible to determine the difference in cost or difficulty of construction of the two models, as the education of the drafting room and construction forces was almost entirely accomplished during the construction of the first one. It is possible that in smaller installations, where the water passages were too small to be readily accessible, the problem of removing the forms from the concrete might assume some importance.

The engineers of the present Niagara Falls Power Co. are glad to certify that the devices adopted and installed have proven equally satisfactory in operation and have enabled each manufacturer to obtain the excellent efficiency of 93 per cent on each water wheel, and the engineers wish to express their appreciation of the great amount of research work done along the lines of regaining the energy contained in the water discharged from the water-wheel runner for useful effect upon the water wheel, and the future design of hydroelectric plants, having in view maximum efficiency results, must embody some form of regainer which will return for useful effect the maximum amount of energy discharged from the water-wheel runner.

H. Birchard Taylor. Mr. White confirms the writer's conclusions arrived at a number of years ago, namely, that in turbines of high specific speed, or during part-gate operation in low-specific-speed turbines, poor conditions of flow will exist in any form of draft tube which is not symmetrical with respect to the axis of the turbine, and although he lays no particular stress on the subject of whirl, which has proved to be the most important element to contend with in any type of tube, his result agrees with the writer's conclusion that the elbow type of draft tube is not the proper type to use in conjunction with runners of medium or high specific speed.

The theories of draft-tube design as developed by a number of European engineers are based on the assumption that there is stream-line flow, more or less parallel to the axis of the draft tube, and that this holds irrespective of whether the axis is a straight line or a curve of short radius, such as is found in the tubes of the elbow type; and American designers in past years have paid heavy penalties for following these theories and thereby ignoring the controlling factor in draft-tube flow, namely, whirl. It would appear from the paper in question that the author has reasoned along

much the same lines, for otherwise one is unable to account for his attempt to establish an analogy between the flow in a free jet impinging on a flat plate and the flow in a draft tube. We have only to admit that a substantial whirl exists under at least some of the operating conditions of most turbines, and under all conditions in many, to conclude that a free jet impinging on a plate offers no proper basis for designing a draft tube which will recover to the greatest degree all of the energy. It must be evident to anyone that if a substantial whirl is introduced into a free jet, the jet disperses into spray, and Mr. White's analogy immediately vanishes.

In order to secure the best results in a draft tube, the writer considers that the shape of the tube should meet three requirements, as follows:

1 The axis of the tube should be a straight line coincident with the axis of the turbine

2 The areas through which the water flows from the runner to the tailrace or collector should be enclosed between surfaces of revolution about this axis

3 Both the axial and whirl components of velocity should be gradually and continuously decelerated. The gradual deceleration of the axial velocity component can be accomplished by a gradual enlargement of the area normal to this component; and the whirl component can be similarly decelerated by gradually guiding the flow to greater radial distances from the axis.

The water is delivered by the runner to the draft tube in a rotating mass symmetrically placed with respect to the axis of the runner; and any form of draft tube having walls laid out on the assumption that it is possible to alter the axis of rotation will produce turbulence, the extent of this turbulence depending upon the extent that the requirements 1 and 2 above are disregarded. As the energy of rotation of the water in a turbine at all gates is relatively great in high-specific-speed turbines, the application of an elbow tube of sharp curvature violates these two requirements to such an extent that the water as a whole breaks into a violent eddy and the possibility of recovering any of the energy in the flow at the discharge from the runner is not only removed, but this eddy imposes a heavy loss in addition.

The writer would call attention at this point to the author's statement in Par. 58; " These comparative efficiencies are based on

the wheels operating at their best speed and best efficiency, that is, when the discharge from the runner is nearly axial."

Pitot-tube and direction-vane tests on high-speed runners have shown that it is a characteristic of such wheels that at increasing gate openings the flow near the axis develops a reverse whirl while the flow near the periphery is still whirling in the same direction as the runner, and that the flow is not even approximately axial at the point of best efficiency.

The draft tube recommended by the author satisfies requirements 1 and 2, but it is in respect to the third requirement that the

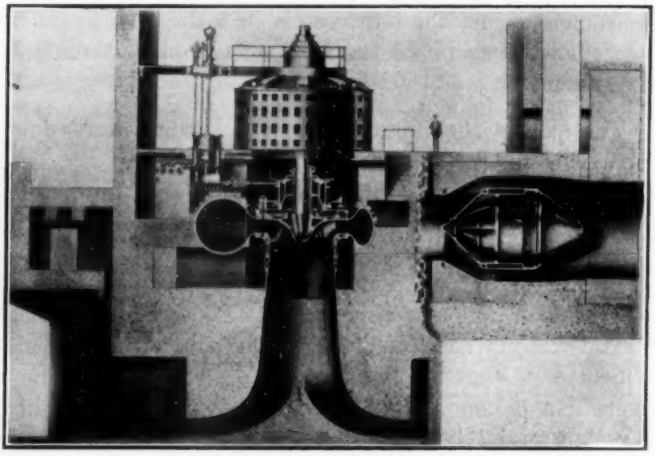


FIG. 32 HYDRAULIC TURBINE FITTED WITH MOODY SPREADING DRAFT TUBE

author and the writer will fail to agree. The author's design of tube which he has attempted to explain by what he calls the "hydraucone action of water" not only presents to the axial components of flow a sudden change of section, but calls for a retardation of flow and then a speeding up, and a final retardation of these components whch are suddenly turned into the radial direction; a condition which invites the formation of eddies and local disturbances, and in water already containing whirl, involves, the writer believes, very complex and unstable conditions of flow.

In Par. 61, the author avoids mentioning the fact that the draft tubes adopted for Units 17 and 18 in the plant of the Niagara Falls Power Company are of the Moody spreading type built by the Power Company from designs prepared by Mr. Moody. See Fig. 32.

The author states that in the case of models of these tubes tested at Niagara Falls the same results were secured both with and without the central cone, and the reader is supposed to infer from this statement that when the cone is removed from the Moody tube we will have the equivalent of the White tube.

The author fails to state that these tests referred to involve the attaching of small models on the end of a supply pipe in which water is received without whirl, and therefore, without duplicating the actual conditions of flow involved in a draft tube applied to a turbine. The author, throughout his paper, has assumed that the comparative efficiency secured on models of draft tubes under conditions of non-whirling stream-line flow would apply unchanged in draft tubes receiving the discharge from runners under actual conditions of operation, and a careful analysis of the paper will show that the structure of the author's argument is built up largely on a series of unrelated suppositions.

The writer desires to call attention to the fact that the tests of the models at Niagara, and likewise the tests on small models described in the author's paper, are comparative tests of diffusers, and a diffuser is not a draft tube until it is attached to a turbine and subjected to the peculiar flow conditions at the discharge of a runner. The difference is vital and the writer is somewhat surprised that the author should not only treat lightly, but practically ignore, differences which are fundamental, and as a matter of fact controlling.

The Moody spreading tube involves the selection of certain area relations maintained throughout the draft tube to satisfy condition 3. The mere placing of a cone in the discharge of a draft tube does not satisfy condition 3.

A large model of a Moody tube used at Niagara Falls was sent to the Holyoke testing flume and tested in a turbine of high specific speed, in other words, under conditions involving whirl, and a drop of about 1 per cent in the efficiency of the turbine resulted when the proper sequence of areas was destroyed by the removal of the cone. In turbines of low specific speed, such as the Niagara turbines, a smaller difference would result.

The author after mentioning Units 17 and 18 at this plant states that "the use of the new substitutes for curved draft tubes at this plant resulted in a water-wheel efficiency of 93 per cent being obtained, which is corroborative evidence of the value of the hydraucone principle for the regaining of pressure from velocity."

It should be recorded at this point that the test results on Units 17 and 18 have no bearing on the "hydraucone principle" which was not used in the design of the spreading draft tubes of these units.

The maximum efficiency of 93 per cent, mentioned in Par. 61, was the result of many factors and represents the combined efficiency of turbine casing, guide vanes, runner and draft tube, and therefore the above figure does not provide any means of separating out the draft-tube efficiency.

The part-gate efficiencies of Units 17 and 18, which are equipped with Moody tubes, are considerably higher than the corresponding efficiencies of Unit 16, equipped with a White tube, the

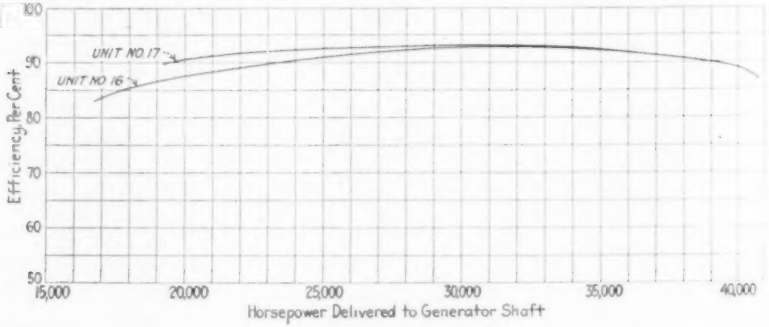


FIG. 33 EFFICIENCIES OF UNITS 16 AND 18, NIAGARA FALLS POWER COMPANY

difference at half-gate amounting to 5 per cent, and although a part of this difference can be accounted for by the fact that the runner used in Unit 16 was designed for a somewhat higher power capacity than the runners in Units 17 and 18, this difference may be allowed for by a relative shifting of the efficiency curves, bringing the results together at high loads and still leaving a difference in favor of the Moody tube of approximately 3 per cent at half-load, as shown by the curves in Fig. 33. Although no data are available to determine how much of this difference is due to the draft tubes alone, it may be stated that the results secured in Units 17 and 18 at Niagara confirm those secured in many series of tests conducted in the I. P. Morris Laboratory. In each series many kinds of draft tubes were used, with the other elements of the experimental turbine remaining the same, and a different runner was used in each series, so as to ascertain the effects

of various specific speeds. In each series of tests the Moody tube demonstrated that it recovers the energy of whirl to a greater extent than any other type of tube. Moreover, Units 17 and 18 of the Niagara Falls Power Co. have been found to be entirely free from vibration or water hammer due to draft-tube disturbances at all gates.

Referring to Table 2 of the author's paper, giving his estimates of the increase in efficiency of power plants by the use of the hydraucone, an important consideration affecting the efficiency has been omitted, the assumptions given in columns 8 and 10 are extremely questionable and the results found in column 12 are incorrect.

It will be noted that in column 7 the author gives the energy of the vertical component of the discharge velocity from the runner, and the writer is able to check these figures from his own experience. The total energy, however, delivered to the draft tube is not only the energy represented in the vertical component, but the energy of the whirl component as well, and the energy which we must deal with, therefore, is the sum of the two. In the case of runner 2, the writer finds that the total energy to be contended with in the draft tube would be about 22, instead of 13.7 given in column 7, and the total energy for runner 3 would be about 30, as compared with 22.4 in column 7.

If the last column is corrected to correspond to these revised values, the increase of efficiency to be secured by using a hydraucone would be 11 per cent for runner 2 and 15 per cent for runner 3. Efficiences of over 90 per cent have been secured in turbines having the same specific speed as runner 2 and equipped with elbow-type draft tubes. If to 90 per cent efficiency is added the increase of 11 per cent as obtained from corrected column 12, we would be led to expect an efficiency of 101 per cent by the use of the hydraucone.

The explanation is probably in the fact that the figures in column 8, representing the efficiency of curved draft tubes, are too low; and that we have no evidence to show that the hydraucone will give an efficiency of 70 per cent when operating on whirling water discharged from a runner. This value of 70 per cent appears to have been taken from tests of the hydraucone when not connected to a turbine. The efficiency of the elbow-type tubes in column 8 might easily be 20 per cent higher than the values given. For example, with runner 1, which is of the same specific speed as

the 37,500-hp. Niagara units, a curved draft tube of 60 per cent efficiency could be used, giving a possible increase by the use of the hydraucone of 0.4 per cent instead of 1.2 per cent as given in column 12.

In conclusion: A free jet impinging on a flat plate or any other kind of a plate involving in the jet pure stream-line flow is a very difficult condition to secure. It is a product of the laboratory and cannot even be secured there except under ideal conditions from a smooth orifice or nozzel attached to a reservoir. Complete freedom from vibration, whirls and eddy currents is essential to its existence, and the slightest jar may immediately destroy the stream-line conditions.

Although the author has demonstrated that by the use of a draft tube satisfying the conditions of symmetry it is possible to secure better results than in any form of draft tube which seriously violates these conditions, it must be clearly borne in mind that it is possible to design various types of draft tubes which satisfy these conditions without properly regaining all components of the entering velocity. It is in the means adopted for decelerating the velocity that Mr. White and Mr. Moody are fundamentally apart.

Lewis F. Moody. The paper contains much interesting and valuable material relating to draft tubes of hydraulic turbines, and the large amount of experimentation which it describes is notable.

Although the forms of tube described include a number of tubes of both the symmterical and elbow types which evidently do not fall within the field of the hydraucone regainer, the writer will confine his discussion to the latter form of tube, which is the chief subject-matter of the paper.

In attempting to analyze Mr. White's explanation of the development of his form of tube, we note that it involves, first, (Par. 2) "causing the stream flow to impinge upon some definite shape, either flat, conical or concave, thus changing its direction, and then placing an envelope around the shape so formed upon the particular base used, which envelope conforms to the shape of the fluid at entrance and gradually recedes from what would be the normal or free shape of the non-enclosed fluid impinging upon the particular base used." In spite of the occurence of the word "definite" in the above quotation, there is so far nothing definite either as to the form of surface against which the

flow is to impinge, or the manner in which the envelope is to "recede" from the fluid surface.

The second element involved is the selection of a flat plate as the base surface and the discarding of various coned and cupped surfaces which were investigated. As reasons for choosing the flat plate the author mentions the avoidance of surface friction on a central cone, and the turning of the water with minimum radius of curvature. He uses the term "hydraucone action of water" to denote the "action which occurs when an unenclosed stream impinges against, and is deflected along, a surface," and he mentions as a part of this "hydraucone action" that "the water at the center of the jet slows up as it reaches the plate and delivers its velocity head into pressure head at the center of the plate. This pressure head at the center is transformed into velocity as this stream flow passes from the center of the point of impingment to the point beyond the radius of curvature."

It may fairly be concluded from the above that this "hydraucone action" involves as distinguishing elements the *impinging* of water *against* the surface (not merely the guiding of water along and parallel to a surface), the substantially complete *extinction of velocity head* at some point in the stream, and the reconversion of the resulting *pressure head* into *velocity head.* If the above does not fairly characterize the "hydraucone action," then the writer does not know of any criterion which would distinguish it from an intangible unreality. Evidently a surface nearly, if not precisely, normal to the flow at the point where the water meets it is essential to this action.

If an engineer unfamiliar with the history of turbine design were to examine the form of tube proposed by Mr. White, in which the discharge from the runner is thus made to impinge perpendicularly against a flat plate and is then made to turn abruptly at right angles and to pass off on all sides, he would naturally be prompted to ask why a flat surface should be interposed in the flow, and why such great efforts should be expended to find the exact form of tube necessary to permit the water to be handled in this particular manner and to avoid undue losses in so doing. Why not turn the water gradually along smooth curves?

The question does not appear to be very satisfactorily answered by anything in the paper, but probably one reason for adopting an arrangement which will turn the water as suddenly as possible is the desire to contract the axial dimensions of the tube to the

extreme limit. Such drastic means are, however, not necessary, since the water can readily be turned along smooth and easy curves within a structure of as small axial extent as is practically required except under very unusual circumstances.

There is, however, a well-established precedent for discharging the outflow from the runner against a flat surface, as shown by the history of turbine development. This may be recognized in what was at one time the common and accepted practice of equipping turbines with straight vertical draft tubes discharging into tailrace chambers having flat floors. Prásil, in Switzerland, had worked upon the same basis, and had developed a tube in accordance with the mathematical theory of hydrodynamics, involving the assumption of non-turbulent flow and the existence of a velocity potential, and depending upon the use of an arbitrary function for the purpose of turning the water from axial to radial between an outer surface of revolution and a flat bottom surface. The Prásil theory, like the method used by Mr. White, was based on the entrance of the water into the draft tube with steady, stream-like flow.

Although the Prásil tube is seriously defective, it has one advantage over Mr. White's tube, which is that it can be definitely designed and its form can be exactly defined. Prásil does not ask us to determine by experiment the shape of the surface of a moving stream of water impinging upon a bottom of indefinite shape, and then to give the tube walls a form *like* this surface but *different* from it.

To go somewhat further into the matter, the hydraucone regainer, as has been mentioned, by allowing the flow to impinge upon a surface which extends normally across the axis of the tube, meeting this axis at right angles or nearly so, causes the central stream elements to be decelerated to zero velocity, their entire velocity head being converted into pressure head; and after turning at right angles, these central elements of the flow are reaccelerated once more by a reconversion of the pressure head set up at the axis into velocity head. Such an action occurring in a draft tube means that some elements at least in the stream are being accelerated and their pressure head is being converted into velocity head, when the object which the draft tube is intended to accomplish is to decelerate all elements of the stream and to convert the greatest possible proportion of the initial velocity head into pressure head.

Furthermore, the above action presupposes the entrance of the flow into the tube in directions parallel to the axis, and without rotation or "whirl" about the axis. When whirl is present, as it almost invariably is, in the discharge from an actual high-speed runner, the conditions of flow pictured by the author do not exist. Since in ordinary flow at velocities above the critical velocity there are secondary or cross currents which mingle one flow filament with another, and since when freely whirling water approaches the axis it increases in velocity in inverse proportion to the radius, owing to its preserving its moment of momentum, it follows that there will be a strong tendency in a draft tube to form a turbulent central core of water surrounded by a rapidly whirling surface of flow, and thus there will seldom be any stream elements at the axis. Mr. White's reduction of friction by discarding a central cone will in all probability be far overbalanced by leaving a central space not partaking of the stream flow and not filled with "dead water" but with water in a violently disturbed and eddying condition. Refer to his statement in Par. 55: "The friction of the water along the surface of a hard cone center is much greater than the internal friction when the water forms its own cone as shown in Fig. 18." This statement does not present to the writer a very clear or consistent idea. If the water "forms its own cone" when a "hard cone" is not used, where is this cone in Fig. 18? The writer does not find any such cone in Fig. 18 or the accompanying description.

The author, in mentioning the draft tubes of the Station No. 3 extension of the Niagara Falls Power Company makes the implication that the draft tubes of units 17 and 18 at this plant, shown in Fig. 32, employ the hydraucone principle. He states that: "The use of the new substitutes for curved draft tubes at this plant resulted in a water-wheel efficiency of 93 per cent being obtained, which is corroborative evidence of the value of the hydraucone principle for the regaining of pressure from velocity." These units are equipped with spreading draft tubes, and the writer can state without qualification that this form of tube is so shaped that "hydraucone action" of water is entirely eliminated and avoided. There is no impinging of water upon a surface and there is no building up of a high local pressure at any point by decelerating the velocity to zero or a value anywhere near zero, and at no point in the spreading tube is there an *acceleration* of velocity.

The spreading draft tube, developed by the writer, was conceived along diametrically opposite lines to the tubes of Mr. White and Dr. Prásil. In this form of tube the water is received between two concentric surfaces of revolution in a direction tangent to both surfaces, and is *guided along* and not *against* the confining walls, being turned along curves having as large radii of curvature as can be conveniently adopted within the space restrictions of a given plant; and deceleration of the flow is gradual and continuous at every part of the passage. The deceleration of the whirl components of velocity is made gradual and continuous by guiding all parts of the flow away from the axis along the central surface, and the deceleration of the meridian components is defi-

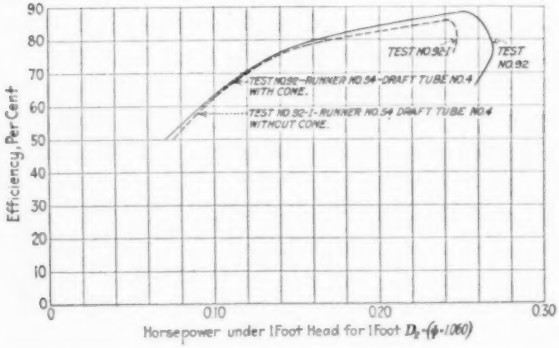


FIG. 34 EFFICIENCIES OF TURBINES, DRAFT-TUBES FITTED WITH AND WITHOUT CENTRAL SPREADING CONE

nitely controlled by the progressive enlargement of the annular cross-sectional area between the two surfaces according to a simple rule.

Mention should be made of a statement contained in Par. 61 of the author's paper. In referring to the model experiments at Niagara Falls, the author states that "The best model with cone center had the proper-shaped conoidal chamber for regain and when the cone center was removed from the model it was found that the efficiency was no lower than with the cone in place." Tests have been recently completed which have a direct bearing on this point. These tests, the results of which are graphically presented in Fig. 34, were made in the I. P. Morris Testing Laboratory on an actual turbine equipped with a spreading draft tube. The turbine was tested both with the central

cone of the draft tube in place and with this cone removed. It will be noted from the curves that the removal of the cone resulted in a decrease of about 2¼ per cent in the maximum efficiency and nearly 9 per cent in the full-load horsepower.

In concluding, the writer would express once more his appreciation of the large amount of research and development work which Mr. White has done in his field, and would also state that he recognizes that the form of draft tube which Mr. White has developed and named "hydraucone regainer" possesses a valuable property in that it is symmetrical about the turbine axis; but the writer feels it to be only fair that he should take exception to the view that all symmetrical draft tubes are "hydraucone regainers," and that if a tube is found to be successful it must, of necessity, employ the "hydraucone principle."

FORREST NAGLER. The writer has been quite familiar with the application of the hydraucone regainers as applied to water wheels, but had no conception of the extensiveness and thoroughness of the work involved in the development of this device. This paper should be of great historical interest and the work outlined should stand as a splendid example of thorough research along a given line.

To the writer, the most evident reason for the use of a device such as the hydraucone is found in the consideration that the water leaving any hydraulic-turbine runner is in a state of rotation, greater or less in amount and clockwise or counter-clockwise in direction, depending on the particular design compared, the gate opening at which it is operating and the head and speed existing at the time. Obviously, the only form which can handle such a variety of rotation must be formed by surfaces of revolution. In other words, it must be concentric with the center line of the shaft, otherwise the natural rotation of water will be interfered with. Hydraulic practice in recent years has demonstrated the extreme necessity for conforming to the natural tendency of flowing water in handling it for a useful purpose, and this applies to the diffuser of a turbine runner more than to any other portion of a hydraulic machine, except perhaps the runner itself.

A long, straight, tapered tube extending far enough to reduce the velocity to an amount which will be permissible as a loss undoubtedly conforms to this requirement. This fact has long been recognized in practice, if not in theory, by all builders, who have found that to obtain high runner efficiency the use of such a tube is essential at Holyoke. Applications of these same runners or

larger sizes of the same type to power-house settings unfortunately have not been accompanied by any adherence to the use of such ideal diffusers. This is evidenced by an inspection of power-plant settings in America where the art of turbine building has undoubtedly reached the highest state of any place in the world. We find all sorts of curved tubes, short tubes, quarter-turn bends, and center-discharge casings and it is remarkable that hydraulic-turbine practice has progressed so far with such considerable handicaps.

The logical reason for not using the long straight tube is found, of course, in the excessive excavation that would be required in vertical plants and the excessive length that would be necessitated in horizontal plants. To overcome these two objections it is necessary to deflect the water at right angles to the shaft within a reasonable distance from the runner. A spiral would serve if the angle and velocity of swirl leaving the runner were fixed, but this is not the case.

To conform to the concentric limitations outlined above and to the limitations of length we immediately run into a form approximating that of the hydraucone.

To the writer's mind, the remarkable feature of the hydraucone is found in the fact that the water can be turned abruptly at right angles in the conoidal chamber without appreciable loss of efficiency. Since that fact has been established, it is easy to find confirmation of this fact in various technical references to the possibility of so turning a jet. Applying such reference to hydraulic-turbine practice, however, met with the usual delay in overcoming any established engineering habit.

The writer has been greatly impressed by numerous instances of fairly close approach to the hydraucone principle but where the ultimate idea was not applied. Many tubes discharging against a flat plate have undoubtedly been investigated and numerous studies have been made in this country and abroad on the effect of a flat plate adjacent to the bottom of a vertical draft tube. So far as the writer has been able to determine all of these instances come under one of the following two heads.

1 The work in question was done to overcome what was thought to be a positive disadvantage resulting from the proximity of the bottom of a tail-race, or

2 The work was performed in order to permit of some mechanical arrangement of cylindrical gate at the discharge of a

wheel or in the attempt to apply a radial Boyden diffuser to some form of an axial-flow wheel.

In all of the instances the outwardly turned passage was considered as the diffuser or else there was no real attempt made properly to form the conical portion of this passage to reclaim the velocity with great efficiency. Numerous experiments by Professor Zowski, Professor Allen, and Mr. Uhl, and in the various plants at Lowell, Mass., apparently come under the first heading. Numerous experiments that have been made at Holyoke and cited in the various reports made of work done there, come under the second heading.

The writer does not completely agree with the concluding paragraph of the paper. It cannot be denied, of course, that any disturbances in the water at the exit end of a draft tube represent energy loss. The extent of this loss is often greatly over-estimated, however.

The extent of the disturbances, according to the writer's opinion, is more a function of depth of submergence of the tube or of the gate opening at which the turbine is being operated, than it is an indication of absolute efficiency. This disturbance increases, of course, as the head goes up, other conditions being the same.

It very often occurs that a poor efficiency of draft tube is evidenced by a tremendous disturbance, but it is hardly true that a tremendous disturbance always indicates poor draft tube or poor turbine efficiency. One of the most excellent installations with which the writer is familiar happens to have a very small submergence of draft tube, and it also happens that this plant is operated largely at gate openings beyond that corresponding to least swirl from the runner. The result is a series of waves averaging perhaps one and one-half to two feet in height, intermittent in character and creating the impression of a tremendous loss. The head is less than 30 ft. so that it might be thought that there would be between five and ten per cent loss in energy indicated by the height of the wave in proportion to the total head. Yet this plant shows over 90 per cent turbine efficiency.

The paper in general should be received by the engineering profession, particularly those interested in hydraulic-turbine practice, with interest, as it represents an exhaustive study of a special and very important branch of the art.

L. F. Harza. In Par. 43 of the author's paper, a reduction of maximum efficiency from 78.7 to 68.8 per cent is shown to have re-

sulted from cutting down the diameter of the plates to 18 inches. The diameter before cutting down is not recorded and quantitative analysis of the reason for this reduction is not, therefore, possible.

This large reduction in efficiency has suggested to the writer, however, that the so-called hydraucone is in reality two separate and distinct pieces of apparatus, the hydraucone proper serving as an elbow to deflect the direction of the water and the annular space from there to the edge of the prallel plates serving in a manner similar to the Boyden diffuser.

From the author's Fig. 18, as also from application of Bernoulli's theorem, it is evident that the freely impinging jet, after being deflected into a radial direction, has the same velocity as before it came into the influence of the plate and began to expand. It is also evident that this velocity and the cross section of flow must continue constant and the sheet of water, therefore, reduces in thickness as it spreads out to greater diameter on the plate.

If d = diameter of inlet to the hydraucone

D = diameter of the true hydraucone where the hydraucone radius becomes horizontal or parallel to the lower plate, and

S = spacing of the upper and lower plates at diameter D and outward to the edge of the plates if parallel, the natural condition occurs when

$$\frac{\pi d^2}{4} = \pi DS$$

or

$$S = \frac{d^2}{4D}$$

Thus if $D = 2d$, then $S = d/8$.

In the diffuser, from diameter D outward to the edge of the plates if parallel, the velocity is reduced in inverse proportion to the increase in diameter instead of remainng constant as under unrestrained flow as in Fig. 18.

The writer has attempted to analyze the author's data to see how far from this natural condition the successful hydraucone designs have been able to depart for the regain of pressure but in most cases the necessary fundamental data for such a study have been omitted by the author. Refer again to Par. 43.

With 4 in. inlet diameter and $2\frac{3}{4}$ in. hydraucone radius, the outer diameter of the hydraucone becomes $9\frac{1}{2}$ in. and in flowing to the outer edge of the 18-in. plates, the velocity of the water is,

therefore, nearly cut in half after the hydraucone proper has completely done its work. At a diameter of $9\frac{1}{2}$ in. the flow area is $\pi \times 9.5 \times S$, where S is the spacing of the plates and when equal to the area of the 4–in. throat, $S = 0.42$ in.

This would conform to the natural flow as shown by Fig. 18. The test shows that best efficiency is with $S = 0.75$ in, which would indicate a velocity reduction in the hydraucone of 44 per cent and in the diffuser of 47 per cent of the remainder.

The kinetic energy at the three points is in the ratio of:

4–in. throat, 12.6 sq. in.	kinetic	energy	= 1.00
$9\frac{1}{2}$ in diam. of hydraucone, 22.4 sq. in.	"	"	= 0.317
18–in. diam. of diffuser, 42.4 sq. in.	"	"	= 0.088

Thus of the maximum efficiency of regain 68.8 per cent as shown in Par. 43, not over $\frac{1.00 - 0.317}{1.00 - 0.088} \times 68.8 = 51.6$ per cent, can be correctly credited to the hydraucone.

In the test curves of Fig. 13 and the apparatus shown in Fig. 9, the spacing of the plates is not recorded. The outer diameter of the hydraucone proper is $28\frac{3}{8}$ in. and of the diffuser 54 in. which indicates a regain in the diffuser similar to that in the above analyzed experiment.

These examples are sufficient to indicate that no intelligent interpretation of experimental results is possible without knowledge of the spacing of plates, diameter of the hydraucone proper, D, and the outer diameter of the diffuser plates, and preferably also the pressure at diameter D. It would then be possible to deduct from the total regain that which occurred in the diffuser leaving the true regains and efficiencies of the various models of hydraucones for comparison. Knowing these results, the regain of any width of diffuser element could be readily added.

Without thus subdividing the apparatus it is feared that the reader of this paper might make false deductions, for each change in hydraucone radius changes the diffuser and with it the distribution of the regain, thus obscuring the real effect of the change in radius upon the hydraucone itself.

By bringing up this point, the writer does not wish to criticise the valuable research work of the author. Undoubtedly his own test records contain all the data necessary for intelligent analysis. He would point out, however, that in most cases the

data recorded in this paper are not sufficient. In the application of this invention it would seem necessary to provide for turning the direction of the water by the hydraucone in the most efficient manner, and thereafter retarding its velocity by increasing diameter of diffuser as far as natural conditions and considerations of economy will permit. The great reduction in efficiency by cutting down the plates in Par. 43 without alteration of the hydraucone is sufficient proof that the apparatus must be subdivided for analysis to avoid false conclusions.

ARTHUR M. GREENE, JR. The improvements in draft-tube design have been a matter of importance during the last few years because in the past many of us have assumed that by giving a gradually increasing area to the tube, the water would accommodate itself to the available area with a lowering of the velocity and a regain of pressure. That this action does not take place we have gradually learned from many installations where proper efficiencies have not been obtained or in places where it has been desired to increase capacities.

The presentations in this excellent paper lead one to hope that this matter will receive more consideration and that we may have other papers on this very important subject. For instance in the Station No. 3 of the Niagara Falls Power Company in addition to Unit 16 equipped with the hydraucone, there are two Units 17 and 18 of the same size equipped with Moody spreading type of draft tube. This tube made possible efficiencies at high load similar to the hydraucone and at part-gate opening is reported to have given higher efficiencies than those obtained with the hydraucone, thus showing that the Moody spreading draft tube is better adapted to regain part of the energy of whirl which exists at part gate. The discussion of the theory of these might lead to the reasons for this action.

In Fig. 18 of the author's paper it appears that the velocities recorded indicate a condition of disturbance in that the various streams are moving at different velocities and are therefore moving one over the other and possibly permitting cross flow. It also may be that the eddies set up would cause the distribution of velocities.

Experiments at the Rensselaer Polytechnic Institute on varying angles of divergence in conical draft tubes with small angles indicate a variation across the cross section of the larger sections which does not exist originally.

CHESTER W. LARNER. The paper is not only a very interesting but a particularly timely contribution on this important subject. It has been generally recognized for some time that the familiar bent draft tube is, at best, very inadequate for the purpose intended and in many cases so poor as to be worse than none at all. Draft-tube design has not kept pace with the design of runners and other important elements of turbine construction. This has been due partly to lack of experimental development and partly to fanciful and incorrect theoretical analysis. Some very beautiful draft-tube theories have been evolved and, unfortunately, put into practice, based on assumed conditions of flow in the tube which cannot possibly occur and which anyone with common sense, whose mind is not beguiled in the pursuit of mathematical pastimes, would recognize as impossible.

The whole theory of the bent draft tube has been built up on the hypothesis of symmetrical filamental flow, notwithstanding our knowledge that water never flows around a bend in any such manner, even when it is not whirling. The author's Figs. 20 and 22 show, in a general way, what actually occurs in a bend when the water enters without whirl, although they do not show or suggest the turbulence and eddying of the water at the top of the horizontal leg. These eddies cause a loss of energy far in excess of the loss of the velocity head at exit, which is so greatly augmented by the reduction of the effective area of discharge. This condition is further complicated by the whirl of the water leaving the runner, which tends to continue in the same plane after turning the bend and thus accelerates the forward velocity at one side of the tube and retards it at the other side, even to the point of causing reverse flow in the tube.

The writer dwells on this subject of erroneous theory because it is the principle obstacle which has held back draft-tube development in the past. It is certainly time for us to arrive at a correct understanding of what actually happens in a draft tube under working conditions and, in this respect, it seems to the writer that the author's premises are open to question and that he is falling into the old error of building up a draft-tube theory on the basis of straight-line flow, which practically never exists.

The author claims to use as the basis of his draft-tube design the shape of a free jet impinging on a plate and he says it is necessary to take into consideration the action of this free jet as it strikes the plate if the maximum recovery of velocity head

is to be expected. This statement, in one form or another, is reiterated frequently throughout the paper and it appears obvious that the author regards the analogy to a free jet as of fundamental importance. This treatment is open to criticism for the following reasons:

1 A free jet presupposes straight-line flow, which does not occur in a draft tube. If whirl is introduced into a jet, the jet is instantly dispersed into spray.

2 The object of a draft tube is recovery of velocity head and there is no recovery of velocity head in a free jet. Fig. 18, in which the velocities before and after impingement are alleged to be the same, indicates that the author appreciates this fact and yet, at the same time, he says that this illustration discloses the principle of "the new method of regaining pressure from velocity."

3 The author does not adhere to the free-jet principle in working out his own designs. He begins to taper his draft tube at the top (see Fig. 31) and diverges more and more from the shape of the free jet as he progresses down the draft tube and around the turn. In so doing he utilizes the diffuser principle which is, after all, the only abstract principle actually involved in good draft-tube design. The problems of design are manifold, but efficient recovery of velocity head is purely a question of reducing the velocity with as little loss as possible. The ideal draft tube is one in which the areas normal to the flow are gradually and progressively enlarged and so shaped and disposed that the velocity and distribution of flow conform to the area provided.

The writer cannot see that the action of an impinging jet has anything whatsoever to do with a draft tube unless it be that, as the author states, it determines the shortest turn which the water can make from the vertical to the horizontal direction. If we were concerned with straight-line flow, this point might have some significance, (granting that it is a fact, as it may be, that the loss in turning is less than it would be if a central deflector were utilized) but with whirling flow the principle is lost because the jet is lost. The central filaments no longer travel in a straight line until deflected by the plate but are thrown outward by centrifugal force establishing a cone of eddying water at the center of the plate on which the outer flow is turned.

The author states in Par. 55 that "the friction of the water along the surface of a hard cone center is much greater than the internal friction when the water forms its own cone as shown in

Fig. 18." Fig. 18 does not show any central cone deflecting the water but, passing over this point, since such a cone would undoubtedly exist if the water were whirling, the writer must take issue with the author on the general proposition as to whether there is less energy lost when water is deflected from its course by an eddy or when it is deflected by the solid wall of a passage. Of course, the shape of the passage might itself be such as to form eddies, in which event there might not be much, if any, difference, but, assuming that the passage or wall is correctly shaped, then the loss would be only that due to skin friction, which is small compared to the loss of energy which results from eddies.

The diversion or guiding of flowing water by impingement upon an eddy or the so-called "turning of water upon water" (which always involves eddies) is, generally speaking, the most wasteful of all methods. We see illustrations of this fact on every hand, such, for example, as the hydraulic jump where the water level is raised by an artificial weir in the form of an eddy. In this case the loss of energy is found to be much greater than it would be if the eddy were replaced by a solid weir.

The design of the author's draft tube is undoubtedly correct in so far as it consists of a circular structure symmetrical about a straight axis coincident with the axis of the turbine, but, when he departs from the diffuser principle and neglects the proper arrangement of areas to the extent of permitting eddies to form at the center of the tube, he goes astray.

The author treats in his paper of the use of a cone at the base of the tube but apparently only as a variation in form of the obstacle upon which he insists that the stream must be caused to impinge. He does not appear to have considered the use of a cone to produce an annular diffuser in which the combined velocity of flow and whirl may be gradually and smoothly retarded. This is a vital distinction and one which it is most important to recognize. In fact, if one may use the expression, the annular diffuser is Nature's form of draft tube, because the whirling water forms a central cone and simulates this type of construction even though no solid cone is actually provided.

Of course, it is undoubtedly true that in many cases the difference between these two arrangements might not be especially noticeable. The difference would be most pronounced in the case of wheels of high specific speed where the whirl is greatest.

Every engineer who has had much experience with draft tubes,

particularly under high heads, has undoubtedly been forced to the conclusion that the center of the tube is a very delicate spot. Even with the turbine located at a very moderate elevation above tail water we find, as the head increases, that it becomes a very difficult matter to maintain friendly relations between the water on opposite sides of the center of the tube. The vacuum in the tube is high due to the high velocity of discharge from the runner and the addition of even a little whirl is often sufficient to break the column apart at the center, causing more or less intermittent cavitation with its attendent vibration and water hammer, not to mention erosive effects on the runner and adjacent parts.

GEORGE F. LOVETT.[1] The writer has been in direct charge of ten installations of White's hydraucone regainers, six of which were in redevelopment work and four in original development.

Two of the redevelopment installations were built in 1916 at the Shelburn Plant of the Brown Co., and adopted over a curved draft-tube design to avoid deep excavation in already treacherous surroundings. The installation is of cement concrete design, and under a seventeen-foot working head of water. The four other redevelopment installations are at the Cross Plant of the Brown Co. and of the steel-plate design on horizontal shaft, and a very economical equipment as far as redevelopment goes, for if curved draft-tube design were adopted at this plant a large amount of ledge would have had to be quarried out, and as the plant was putting out partial power during reconstruction, ledge excavation was prohibitive.

Three steel-plate vertical designs are now installed at the new Riverside Plant and again in ledge, and the saving in deep excavation was the governing factor; also one steel-plate vertical design at the La Tuque, P. Q., plant, also in ledge.

The governing factor in all these installations has been the tail-race excavation, and while no official reports on efficiency of the units has been received the company is satisfied that the output of the units is beyond expectations.

EDWARD W. BURBANK. The hydraucone regainer is without doubt one of the epoch-making inventions in the hydraulic turbine field. Its speedy acceptance in its varied forms by the engineering profession attests to the highly efficient results which can be expected from its installation in connection with hydraulic-tur-

[1] Brown Co., Berlin, N. H.

bine settings. The day has passed when large developments are made with a consideration of high runner efficiency only; the efficiency of the complete setting must now be taken account of, and there is no part of a turbine unit which permits of such a decided change in design with such a great increase in efficiency as does that of the energy regainer on the discharge side of the runner, and this change has been effected by the substitution of the hydraucone regainer for the common and accepted type of curved draft tube.

Many repeated tests on hydraucone regainers, with flat bottom plate and with cones of varying heights as well as depressions in the plate, gave always the same result — that of increased efficiency for the regainer with the flat bottom plate over that of any of the other forms. The difference in efficiency with hydraucone regainers of varying types would not be so noticeable in high-head developments such as is Station No. 3 Extension of the Niagara Falls Power Co., but careful testing of the varied forms in connection with similar high-specific-speed runners on low-head work would corroborate the results obtained in the laboratory experiments which were made on no mean scale.

As the design of the hydraucone can be changed as to its radius and as to the volume of its conoidal chamber, the latter, principally by the placing of various sizes of cones at the center of the bottom plate, it is not difficult to see that certain regainers which have been recently constructed, and which at first the engineering profession thought embodied a totally different principle, are in reality nothing but a simple variation of the hydraucone.

As the writer was a witness to the construction of the experimental apparatus and the execution of the lengthy and most complete tests on the various working models, from the beginning of the development of the hydraucone regainer to its final and accepted form, he will certify to the care, precision, and correctness which obtained in the making of the various readings, the computation of the figures, the plotting of the desired information, and the authenticity of the results as presented by the author.

Arnold Pfau. Referring to Fig. 18, is there a more convincing proof that Nature is solving its problems in the most perfect manner? A column of water is directed upon a flat plate at a certain velocity and is deflected at right angles, at the same time the energy contained in the water is gradually transformed. Here is the hydraucone regainer, the draft tube prescribed by Nature,

the most perfect solution of the problem, accomplishing everything that should be accomplished, namely:

1 Transforming the static suction head into pressure, thus restoring head otherwise lost

2 Transforming the kinetic energy of the discharging water into pressure, again restoring a part of the head otherwise lost

3 Transforming into actual head such energies as are latent in the disturbances, resulting from an incorrect discharge of the water from the runner, such as rotating components and whirls.

It must be conceded that no matter how far the plate is moved toward the orifice of the jet, the hydraucone discharge represents the best solution of the problem as presented. The distance of the plate from the orifice seriously affects the cost of excavation of a plant. Therefore, the saving in cost can be balanced against the sacrifice of efficiency of the draft tube with the full conviction that the problem as it is presented is solved in the most efficient possible manner.

Joseph J. Ring. During the series of experiments a large number of small draft tubes, exact models of those designed by the Allis-Chalmers Company and also of other designs for some of the largest installations, were built and tested with the same runner and their comparative efficiencies obtained. While these results will vary some from those obtained with the actual draft tubes, because the characteristics of the runners used varied from that used in the tests, the results cannot vary very much and confirm the conclusion arrived at in the paper by the author that the curved draft tube is a very poor means of regaining the energy discharged from the runner.

During the period of these tests the writer made a wide study into the various devices used for draft tubes in connection with water wheels both in this country and in Europe, and all of these devices and every other conceivable device that anyone could think of were tested out in our 4-in. model tests, and comprehensively covered the field of draft tubes as it was developed up to 1915.

The writer found a number of devices that were in some respects similar to the hydraucone regainer, and had combinations of some of these been made results similar to those obtained by the use of the hydraucone might have been expected, but the writer found nothing that would lead him to believe either in theoretical discussion or in practical application that anyone had grasped

the idea of the hydraucone action of water or its application to the water wheel.

The tests made at the private laboratory were all made with one runner of a specific speed of about 90. In the first series of tests made at the Holyoke Testing Flume a runner with a specific speed of about 75 was used; the next series of tests were made with several Francis runners with varied specific speeds running from 100 to 120, and the last series were made with the new high-speed runners of the type developed by Mr. Nagler with specific speeds varying from 125 to 200. All these tests were made first with the best straight conical draft tubes that the Allis-Chalmers Company had developed especially for high-efficiency tests at Holyoke, and then with the hydraucone regainer leaving all other features the same. The results of these tests show conclusively that the hydraucone regainer gave in all cases just as good results and in most cases better results than did the conical tubes.

The natural conclusion from these tests is that by using the hydraucone regainer with a runner of almost any specific speed, results can be obtained in the commercial power house that will be as good as the best obtained at the Holyoke Testing Flume with the best possible testing equipment used there, without excessive excavations or prohibitive cost in the power house or in the hydraulic equipment used, and that the hydraucone regainer is a long step forward in commericial hydraulic-power development.

J. R. James. The hydraucone has many points in its favor, especially so since tests indicate that its efficiency is as good as, if not better than, the ordinary conception of a draft tube.

Looking at this proposition from the construction angle, any reduction in depth of excavation is certainly a valuable consideration. Naturally, deep excavations are not only very expensive but the element of risk is always that much greater. Furthermore, the conventional type of draft tube involves large quantities of mass concrete.

To decrease still further the depth of excavation, would it not be possible to utilize the space shown in Fig. 24, represented in cross-section as a quadrant of a circle and immediately above the flat plate, as an exit for the water, rather than the space below the flat plate as shown in Fig. 31?

It is rather interesting to note that the addition of a cone on the flat plate tends to reduce efficiency rather than to increase it.

The hydraucone can be made useful in a central-station steam plant also, especially so in handling the water used for condensing the steam from main units. The path of the water in this case is in reality an inverted or vertical syphon, water starting from the forebay or intake canal passing up through the pump and through the condenser tubes and then dropping into the tail-race or overflow canal. Therefore, any method which will change velocity head into pressure head when the water enters the tail-race will help out the circulating motor to just that extent.

GARDNER S. WILLIAMS. In 1899 the writer was called upon to supervise the tests of the hydraulic machinery for the Lake Superior Power Company which was then installing its plant at Sault Ste. Marie, and for the following two years he spent much time in the hydraulic testing flume at Holyoke on an intensive investigation of the performance of water wheels. As a result of the experimental work done there, turbine efficiency was raised from 68 to 83 per cent on the wheels proposed.

These tests were made on turbine units of two wheels on a horizontal shaft, but separate tests were also made of the individual wheels set as in Fig. 27 of the author's paper with the platform removed. Tests with these vertical settings indicated an increased efficiency of about two per cent over those with horizontal settings. Therefore, when in 1905, it became necessary to prepare plans for the installation of the Edison Sault Electric Co. of Sault Ste. Marie, a vertical turbine operating under a head of fourteen feet and running at 100 r.p.m. was adopted. The draft tubes for these turbines were of the curved type, expanding gradually from the outlet of the wheel, being circular at the turn but expanding horizontally beyond the turn. This was the first low-head installation using the vertical direct-connected or umbrella type unit in this country so far as the writer has been able to discover.

Later, the writer had an opportunity to design a plant containing two units for a 14–ft. head, one driving a direct-current and the other an alternating-current generator. The contract called for a guarantee, with bonus and forfeit provisions. The draft tubes were designed by the writer to turn at right angles, in shape approximating the natural course of water flowing around such bends. Upon seeing the proposed design, the contractor preferred using his own, so, as a compromise, one turbine was fitted with the writer's draft tube and the other with the contractor's. Under test, the contractor just met his guarantee

of 83 per cent efficiency, while the efficiency with the writer's draft tube was 88 per cent. The plant was known as the Argo plant of the Eastern Michigan Edison Co., situated at Ann Arbor, Mich., and the draft tube is referred to by the author.

It was as a result of these tests that the author was able to persuade his company to undertake an investigation to determine how this increase in efficiency of 5 per cent might be made. Later, the hydraucone was developed, and the writer learned that better results were obtained from a model of it than from a model of the Argo draft tube. Therefore, when he was called upon to build another plant with two units, he had one constructed with a hydraucone regainer and the other with a draft tube of his own design, based on the stream lines of a jet impinging on a flat plate. Results showed that an efficiency one to two per cent greater was obtained with the writer's draft tube with a range of gate from three-quarters to full, the hydraucone giving 85 per cent and the writer's tube 87 per cent. The writer's draft tube was provided with steel fins projecting into the tube about eight inches and extending six or eight feet below the wheel so that the whirl of the water was considerably checked.

The writer feels entirely warranted in saying that up to the present moment and for moderate heads it is possible to design an elbow draft tube of just as high efficiency as any other type. This is not to say that the elbow type is the most economical or desirable. Much depends on the installation. With the hydraucone, width is necessary, hence it is a question of width against depth with these two types.

Referring to the previous discussion, the writer would say that the suggestion of utilizing the space surrounding the nozzle of the hydraucone has been carried out in a plant constructed after his designs at Cheboygan, Michigan, giving very satisfactory results in the case of a horizontal setting.

J. F. Roberts. Engineers as as whole appear to be becoming more and more commercialized. The majority of us lack the keen enthusiasm of the artist and judge our ability more by our financial rating than by the scientific results accomplished in our daily work. At the end of the day we gladly drop every thought of work and turn for diversion to the lighter things.

New ideas and great improvements which require months of experimenting and hours of thought in the proper interpretation of results cannot be accomplished solely in the daylight hours. Con-

trary to general opinion great improvements are not the result of a flash of the mind but are the outgrowth of concentrated thought and patient experimenting.

When an engineer offers to the public the fully developed results of carefully worked out ideas and experiments, such as the hydraucone, we must unconsciously place its designer and inventor among the higher engineers, whose ideals have led them through hours of painful labor to the perfection and accomplishment of a purpose, not for its pecuniary reward, but for the sake of that ideal.

To Mr. White, we owe sincere gratitude for the development of the hydraucone, which by the universality of its acceptance and reproduction, has been proved to be a new and useful invention.

THE AUTHOR. It is a credit to our engineering profession that Mr. Shepard has had the courage to come forward so frankly, and without fear or favor has told the essential facts leading up to the development of the so-called Moody spreading draft tube. The author is glad to have the company which he represents included as "one of the two leading water-wheel manufacturers of the world."

It is significant that at the time of the conduct of the negotiations in the fall of 1917, according to Mr. Shepard, the Morris Company engineers put forward their most efficient regaining device in the form of a curved draft tube, and the writer remembers the difficulty encountered in convincing Mr. Shepard that the curved form of draft tube was not proper, but that a concentric draft tube embodying the hydraucone was the one which should be adopted on the new units at Niagara Falls. Mr. Shepard states that it was only after tests had been made on the Morris curved tube and the hydraucone that the Morris engineers came forward with other forms of tubes, and it is interesting to note from the records of Mr. Shepard's tests that each tube submitted by the Morris engineers was successively nearer to the hydraucone regainer. The author must take issue with Mr. Moody when he states that the spreading draft tube is fundamentally different from the hydraucone. The author must even take issue with Mr. Moody on the term "spreading draft tube," because that does not properly fit the case. At the time of the author's development of these new regaining devices he was confronted with the same problem no doubt which Mr. Moody found in the selection of a proper name for the device. Many words were considered, "spreading" among them, in an attempt to find a name which would convey the action of the water involved, but it was found necessary to invent the

word " hydraucone " as set forth in the paper. The application of the term " spreading draft tube " to modern hydroelectric power plants might apply more specifically to the curved form of tube in which the side-walls of the horizontal portion of the tubes are very much separated, and the water spread out with the upper water of the tube brought down close to this spread water as in fact one such tube was developed and tested by Mr. Shepard. The word " spreading " might apply to a concentric draft tube if the water spread out with even an equal velocity, but when the water from a draft tube is spread out by a conical surface in line with the axis of the flow, the water changes the direction or " spreads out " it is true, but with this difference, that the area of the water normal to its flow at the point of greatest curvature of spreading out is enlarged, caused by the pressures produced by the centrifugal force of the water in going around the bend as shown in Fig. 18. The amount of this increase of area of the water at the point of greatest curvature is dependent upon the size of the radius of curvature of the deflecting cone with reference to the size of the stream or jet. There is no cone of the greatest dimensions which can be installed in connection with large water wheels and along the surfaces of which the water may be spread out but what will have a curvature sufficient to cause this increase in the area normal to the flow at the point of greatest curvature, and as the cone is decreased in size relative to the jet it approaches nearer and nearer to the fuller hydraucone as shown in Fig. 18. It is this action of the water as it spreads out that the author has termed the hydraucone action of water, and a re-reading of Par. 2 of the paper will no doubt make the fact clear to the reader that one form of the hydraucone regainer consists of conoidal walls shaped gradually to diverge from the surface of the water conoid which would tend to form on the particular conical base used. Mr. Taylor overlooked the dimensions used on the conoidal draft tubes on units 17 and 18 at Niagara Falls when he states, under his condition 3, " the gradual deceleration of the axial velocity component can be accomplished by a gradual enlargement of the area normal to this component." Mr. Moody has gone to considerable pains in one of his disclosures to show that the area normal to the axial flow of the water must not be enlarged gradually but increased over that rate at point of greatest curvature as the water flows down the cone.

Probably this matter will be clearer to the reader if he will

now consider Fig. 18 of the author's paper where it will be noted that the four outer stream lines, for instance, diverge one from the other as the water begins to flow around the curve and gradually flow together again as the water passes off along the plate. In most of the published matter reference has been made to the impinging of a jet upon a flat plate, but the substitution of a cone and the direction of the water around the cone rearranges the stream flow lines but very much like that shown in Fig. 18, and the reactions within the curvature are such that the area of the jet normal to the flow at point of greatest curvature is increased. A rereading of Pars. 30, 31 and 32 of the author's paper should shed light on this point.

The space beneath the turbine in the usual power house is limited and if a concentric form of regainer is used the direction of the water must be changed with relatively short radius of curvature having reference to the diameter of the discharge from the wheel, in any event with such short radius of curvature that the pressure set up in the stream flow due to centrifugal force in the water in passing around the bend does make it necessary to provide for the natural enlargement of the stream at point of greatest curvature. In other words, hydraucone action of water must necessarily be present. The long series of experiments described in the paper have resulted in the development of a hydraucone regainer consisting essentially of conoidal walls concentric about the axis of the runner. The author submits that the words "spreading out" do not fully cover the action which takes place when a conical surface of revolution is placed in the line of flow from a water wheel but that other actions are present. The author cannot understand how Mr. Taylor got the impression that the author did not take whirling action into account. In another disclosure the author has clearly set forth the benefit resulting from radial passages at the discharge of a water-wheel runner wherein the whirling action of the water may be utilized in increasing the efficiency of the wheel, and in fact in Pars. 13, 37, 54 and 58 of this paper, specific mention is made of the whirl and the advantages of the use of radial passages in connection with the Francis type of wheel. In Par. 58 the words "when the discharge from the runner is nearly axial" should manifestly be understood to mean "when the discharge from the runner is as nearly axial as possible," especially in view of the statement made in Par. 13 that "the water discharges from the runner at best efficiency with a decided whirl."

It is common knowledge among water-wheel designers that the water discharged near the periphery of the wheel when it is operating at best efficiency has a whirling action rotating with the wheel, while the water near the center sometimes rotates with the wheel again approximately axially and in some cases is whirling in a direction opposite to the rotation of the wheel.

Whirling action was present in the runner used on the tests referred to in the paper and this influenced the development of the hydraucone regainer, and the knowledge of the whirl was one of the reasons for having radial passages extending outward from the runner wherein the whirling action of the water could be utilized for increasing the efficiency of the wheel. Referring again to Mr. Taylor's three requirements, the hydraucone regainer complies with the first two of them, but the so-called Moody spreading draft tube does not comply with the third because the actual dimensions of the cone and enclosing envelope are made to provide for that enlargement due to centrifugal force, which is one of the actions which the author has grouped under the general headng Hydraucone Action of Water. Mr. Taylor states, "the difference is vital and the writer is somewhat surprised that the author should not only treat lightly, but practically ignore, differences which are fundamental, and which are as a matter of fact controlling." To controvert such a statement as this it is only necessary to state that one of the hydraucone regainers with cone center which was tested with the turbine at the laboratory conformed practically line for line, cone and all, with the so-called Moody spreading draft tube installed at Niagara Falls.

It should be apparent to any engineer having a reasonable knowledge of hydraulics, that the flow of water does not take place in smooth stream lines, but we sometimes use that term as a basis for theoretical discussion. The author reiterates his statement that the so-called Moody draft tubes at Niagara Falls are mere redevelopment and are actual duplicates of models tested by the writer, and the action within the Moody spreading draft tube comes well within the meaning of the author's claims. The author takes vigorous exception to Mr. Taylor's statement that "it is in the means adopted for decelerating the velocity that Mr. White and Mr. Moody are fundamentally apart." In Mr. Moody's disclosure elsewhere he referred to the enlargement of the cross-sectional area of the stream flow at the point of greatest curvature as

it passes down the cone and uses one method of arriving at the normal free surface of the conoidal water flowing over the cone and then places around this water conoid an envelope of gradually increasing greater capacity than that to just enclose the conoid. A mere difference of words and a mere different method of arriving at the same physical object, which had already been created, does not make that object a different thing. The whole purpose of Mr. Taylor's and Mr. Moody's discussion apparently is to confuse the issue and lead the reader to believe that hydraucone action of water is solely the impingement of a jet upon a flat plate. A careful re-reading of the author's paper will show the fallacy of their arguments.

In Mr. Moody's discussion he quotes from Par. 2 of the paper and says, "in spite of the occurrence of the word 'definite' in the above quotation, there is so far nothing definite either as to the form of surface against which the flow is to impinge, or the manner in which the envelope is to 'recede' from the fluid surface." Mr. Moody's understanding of the word "definite" is not what the author intended to convey. The word "fixed" would be better. It does not matter, as has been frequently stated, as to just what the shape against which the water flows may be; preferably it should be concentric with the axis of the inflowing stream. The water does form itself in some reasonably fixed shape depending upon the base used, upon the "stream lines of flow," if one may use such words, of incoming water; that is to say, the water may be whirling in one direction around the periphery of the stream and whirling in the opposite direction around the center, but it will tend to take some form upon the particular base used. Mr. Moody has outlined elsewhere one method of determining such shape, if we grant certain assumptions which he has made, but what this paper does point out clearly is that the shape is formed which may be used for regaining pressure from velocity by gradually diverging the walls around such shape away from what would be the normal free flow and at such rate as to slow up the water and regain pressure from velocity similar to that action in the discharge end of a venturi meter. The fluid surface of a free jet impinging on such a cone cannot be computed by assuming equal velocity at all points in the stream flow as it flows down the cone, but the action of centrifugal force due to the water turning at the bend must be taken into account which tends to thicken the stream flow at the point of greatest curvature, which thickening is one of the pheno-

mena which the author has included under the term hydraucone action of water. The manner in which the envelope is to recede from the fluid surface is as set forth in Par. 2, " the angle of divergence of the envelope from the normal or free shape of the impinging fluid is such as to bring about the same phenomena of flow condition as result in an expanding straight-axis tube such as the discharge end of a venturi meter." It is well known and is a matter of publication for many years that this angle of divergence may be 4 deg. or reasonably greater, depending upon the space available and the degree of regain desired. Referring to the fifth paragraph of Mr. Moody's discussion, the author feels that he must reiterate again that the words hydraucone and hydraucone action of water are meant to cover those actions, changes, variations in velocities, variations in pressure, thickening and thinning of the stream flow which occur when a stream of water is deflected by a flat, conical or convex surface and does not necessarily imply the extinction of velocity at some point in the stream and the reconversion of resulting pressure head into velocity head. This latter occurs only when the jet impinges upon a flat surface or upon some concave surface. Mr. Moody states, " Why not turn the water gradually along smooth curves?" The introduction of a cone in the stream flow such as was used on Units 17 and 18 does not conduct the water along a sufficiently smooth curve to avoid the hydraucone action of water, which in that particular case is one of the features for the thickening up of the stream flow at a point of maximum curvature of the stream as it flows around the bend. The fact that the cone introduced in the conoid chambers at Niagara Falls is not sufficient to avoid the hydraucone action of water is shown clearly by the results of Mr. Shepard's experiments because when he removed the cone from the Moody model it was found that no loss in efficiency of the device as a regainer resulted. In other words, the cone is mostly camouflage in this case. As it has been pointed out so frequently and is evident from the plotted velocities in Fig. 18, the center of the impinging jet is more or less inactive. It is the proper shaping of the outer conoidal chamber which is most instrumental in affecting the regain so valuable to water-power installations. Mr. Moody makes reference to Prásil's theory but that theory has reference to the shape of the tube without reference to the impinging surface and is in no way similar to the author's discovery. Mr. Moody is hardly consistent when he elaborately sets forth in his tenth paragraph his objections to the

deceleration and reacceleration of the water, when at the same time in one of his recent disclosures he explains how centrifugal force causes certain of the filaments to slow up at the point of greatest curvature along the cone center and reaccelerate again, as the flow passes off the bottom of the cone.

This simply means that hydraucone action is present when the water flows around a cone such as Mr. Moody describes and if the cone be reduced in size hydraucone action increases, and when the cone has been reduced to the limit, namely, a flat plate, greater hydraucone action is present. Mr. Shepard's tests have clearly shown that there is no particular difference between the particular form of the so-called Moody draft tube to distinguish it in shape or performance from that of the hydraucone.

The terms used by Mr. Moody and the author to describe their conceptions of the actions in the two tubes differ far more than the devices themselves which are designed for identically the same purpose, accomplish the same results, give the same high efficiency, the only difference being that the author's device was conceived, developed and put into operation prior to Mr. Moody's. It is interesting to note that Mr. Moody has recently conducted tests of a conoidal chamber with conical center and without conical center, both in connection with a water wheel, and has found with that particular conoid and with that particular cone a difference in efficiency of $2\frac{1}{2}$ per cent. Not having the dimensions of the conoid and cone referred to, the author is unable to point out the probable cause of this difference, but the experiments of the author herein set forth show little difference with and without the cone.

It should be noted that each water-wheel runner has its own peculiar whirling discharge, usually positive about the periphery, negative near the center, all of which affects the results when tested under any particular conditions such as a conoidal chamber with and without cone.

Mr. Moody says, "There is, however, a well-established precedent for discharging the outflow from the runner against a flat surface, as shown by the history of turbine development." Mr. Moody is correct, and his statement tends to confirm the author's reiterations that the hydraucone action referred to is not so much the center of the point of impingement as it is the fact that the water does form a sufficiently definite shape, whether or not it be whirl such as comes from the usual water-wheel runner to enable a properly shaped conoidal chamber, placed around such

normal shape, to regain pressure from velocity, with and without whirl, resulting in the greater efficiency of the turbine, which is the end sought.

Mr. Larner says, "the author claims to use as the basis of his draft-tube design the shape of a free jet impinging on a plate and he says it is necessary to take into consideration the action of this free jet as it strikes the plate if the maximum recovery of velocity head is to be expected." If one will substitute for the word "plate" used above, "either plate, cone or other concentric form," then Mr. Larner's statement is substantially correct. The consideration of the actions within a free jet impinging upon the particular base will lead to a determination of the surface of the water conoid which would tend to form on the particular base used, and then the placing of an envelope around this conoid gradually diverging from the shape it would take when a conoidal chamber was not used secures the ideal draft tube. Mr. Larner says, "The ideal draft tube is one in which the areas normal to the flow are gradually and progressively enlarged and so shaped and disposed that the velocity and distribution of flow conform to the area provided." The words "gradually" and "progressively" are to be understood to refer not to the actual cross-sectional area, as Mr. Larner seems to mean, but to the additional area necessary to secure proper regain as the water flows around the bend. It is to be regretted that Mr. Larner did not sufficiently study the paper to see that the author not "only considered the use of the cone to produce an efficient diffuser in which the combined velocity of flow and whirl may be gradually and fundamentally retarded" but did actually conceive, design, build, test, record and publish the results of a water-wheel operating under those conditions, and also the results with a conical chamber with the cone and with the cone omitted, and the results were printed in this paper for inspection, and show that with the particular type of runner used higher efficiencies were obtained with the cone left out. It is possible that other runners and particularly runners of other specific speeds will give better efficiencies with hydraucone regainers having cone centers than with those not having cone centers; but it should be manifest to any careful reader of the paper that the author has a right to claim, as he does, not only the conoidal chamber without the cone but the one with the cone, in both of which types there are annular passages in which the whirling action of the water is regained for useful effect upon the turbine. Mr. Larner is correct

when he says that draft-tube design has not kept pace with turbine design, but the author has presented a series of tests covering many forms of tubes, particularly many concentric forms in which there are annular passages in which the whirling action of the water is beneficially expanded for increase of efficiency of the turbine.

The greater portion of Mr. Larner's discussion, however, is based on theory, and he does not take fully into consideration the actual facts and test results presented by the author showing that the conoidal chamber with concave bottom, flat bottom, and cone bottom does efficiently regain pressure from the velocity discharged from the type of runner as shown in the paper.

The Moody draft tube is only one of the many types tested by the author.

The author is glad to have Mr. Lovett's statements concerning the construction of the hydraucones in various power plants.

No. 1795

ADVANTAGES OF LARGE FREIGHT LOCOMOTIVES, PARTICULARLY THE 2-10-2 TYPE

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In this paper the author points out the more important considerations involved in a study of the economic value of various types of motive power and demonstrates the complexity of the problem. Probably, he holds, no absolutely correct analysis is possible, and surely not practicable. The choice of motive power is of extreme importance because the characteristics of the power affect the earnings more than any other single factor and determine the efficiency of operation usually throughout the life of the engine. For this reason the choice of the locomotive should be made with extreme care. The final decision should be based on a definite knowledge of the economies that can be realized, not on unsupported opinion. Engineering methods are essential in working out the solution, and the study of the problem offers a field for constructive coöperative work by the members of the engineering societies.

PROBABLY the briefest presentation of the advantages of large locomotives is that made by James J. Hill: "Receipts are by the ton- and passenger-mile; expenses are by the train-mile." It is not to be expected that such a general statement will hold good in all particulars, nevertheless it is true that a large proportion of operating expenses decrease as the trainload increases, although not in the same proportion as the decrease in train-miles. Under the operating conditions existing on most of the main-line mileage of this country, the greatest possibilities for economy are probably still to be found in the adoption of locomotives of high capacity.

2 The large locomotive, designed merely for high rated tractive effort, is not a panacea for operating troubles. The first requisite is a design suited to the conditions of the operating territory, the traffic, and the service. The relative advantages of specific designs are a problem in the economics of operation that must be solved by the application of engineering principles. It is a

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question of adapting the design to the operating and economic conditions and then coördinating the motive power with other facilities. The last requirement is of great importance, for unless all the varied operations can be kept in step the machine as a whole cannot run smoothly and efficiently.

3 The adoption of improved power should be only one part of a coördinated program. Every appropriation for larger engines should carry with it, as an integral part, provision for facilities to insure the maximum utilization of the power. Engine terminals, shops, yards, the rolling stock and the track structure itself should be prepared to assist in obtaining the proper operating results. The coördination of facilities deserves careful study. Much better results will be obtained by the use of locomotives adapted to existing conditions where the related facilities cannot be adapted to the locomotive.

4 The choice of motive power is of extreme importance because the characteristics of the power affect the earnings more than any other single factor and determine the efficiency of operation usually throughout the life of the engine. The problem of introducing new locomotives is similar to the problem of reducing grades and should be studied as thoroughly. The interrelation between the various factors affected affords large opportunities for savings and also for losses. No executive should be satisfied with a superficial analysis of the probable effect of a new type of power on operating costs. This is one of the problems of operation that has often been studied in a general way, but has seldom been analyzed quantitatively. The author has made an attempt to develop a general method of determining the savings that might be expected from the use of high-capacity locomotives, but at every step was so hampered by lack of accurate information that the conclusion was finally reached that there is no method of solving the problem without extensive research. Many investigations have been held up during the past few years and the rapid changes in prices have made earlier data inapplicable. It would seem, therefore, that there is a large field for research in the economic problems of operation when normal conditions are restored.

5 The operating statistics as compiled at present are valuable for the analysis of existing conditions, but they furnish little information as to what would happen under other conditions. Thus the predetermination of operating results, which is most important from the standpoint of improved operation, is

largely dependent upon the researches of the individual roads and of engineering organizations. Vast numbers of unrelated figures are compiled, but they are of little benefit in the solution of the general problem. What is needed is not merely a statement of the expenses under fixed conditions, but the rate of change of expenses under certain varying conditions, such as trainload and speed. If such data were available, the analysis of operating results could be made more truly a diagnosis, rather than a mere post mortem. The investigation referred to above brought out numerous aspects of the economics of operation that apparently merit attention and a few comments on these points may be pertinent.

6 Most of the available reports on the economic value of heavy locomotives consider comparatively few of the items affected. The comparative costs of wages of train crews, of fuel and water, and of repairs to locomotives are often the only items considered. In some few cases the comparative mileage and fixed charges on the investment in motive power have been computed. This is not sufficient to determine conclusively the relative merits of various types of power. The locomotive has a direct or indirect influence on many items of expense in the maintenance of way, maintenance of equipment, and transportation accounts. The real problem in determining the value of a locomotive is to find the effect that its operation will have on the sum total of these accounts. The complexity of the problem has apparently often deterred railroads from giving it detailed consideration. However, it is a matter that goes to the very heart of the problem of economical operation, and the results of a thorough study should more than justify the labor involved.

THE EFFECT OF MOTIVE POWER ON MAINTENANCE-OF-WAY EXPENSES

7 It is the purpose of the following paragraphs to point out some aspects of the problem that apparently are deserving of attention. A search through the literature on this subject has failed to disclose fundamental data on these questions that are applicable to present conditions.

8 Some roads have reached adverse decisions on the adoption of 2–10–2 type locomotives on the ground that the increased cost of roadway maintenance resulting from their use would more

than offset the savings in wages. This opinion does not seem to be generally held, but as one of the arguments against heavy locomotives it deserves recognition. Maintenance-of-way expenses make up about 17 per cent of the total operating expenses, but the greater part of the expenditures are independent of the character of the power. Much of the work of track maintenance is made necessary by the action of the elements, or by the necessity of maintaining the permanent way in suitable condition for fast passenger traffic. The expense which is most directly affected by heavy locomotives with long, rigid wheelbases is rail renewals. Other accounts that are affected to a lesser degree are ties, track laying and surfacing, roadway maintenance, and superintendence.

9 There is little or no information available as to the comparative effect of four and five pairs of coupled wheels on rail wear and the other accounts affected. The more rapid wear of tires indicates that the effect on the rail is appreciable and the tendency to straighten out the track no doubt increases the cost of maintaining it in line. The actual effect will vary according to the wheelbase, the curvature of the road, and whether the locomotive has one or more pairs of drivers equipped with lateral-motion devices.

10 The sum of the maintenance-of-way expenses which may be increased by heavy motive power is about 10 per cent of the total operating expenses, and if the effect is to increase these items considerably the saving will be difficult to make up in other accounts. However, if the wear and tear on the track is merely proportional to the weight of the engine, as is sometimes assumed, light and heavy engines would be on a par as regards these items. The difference of opinion on this question suggests the necessity for a careful investigation.

MAINTENANCE OF EQUIPMENT

11 In any study of locomotive operation the cost of equipment maintenance deserves careful attention. The percentage of the total operating expenses falling in this classification has shown a fairly consistent increase over a considerable period. Locomotive repairs and renewals, which in 1898 amounted to 5.9 per cent, in 1918 had increased to 11.7 per cent. So many factors may influence this ratio that no definite conclusions can be drawn, but it is significant nevertheless.

12 The principal difficulties in maintaining large locomotives

are due to the short life of driving-wheel tires, driving boxes and main-pin bearings. With the proper facilities and proper construction the work of caring for these parts becomes merely a matter of routine running repairs, but where the lack or inadequacy of terminal facilities hampers repairs, the loss of service due to these minor items may become serious. In extreme cases the mileage per month may be reduced so much as to make the fixed charges per ton-mile unreasonably high. It is hardly necessary to point out that this should not be charged against the locomotive itself.

13 While the foregoing remarks are confined to some of the more important items of roundhouse maintenance, they are equally applicable to the work of classified repairs. If the shops and shop machinery are not adequate for new power, repair charges will be high and the time out of service will be increased. The cost of these facilities should be considered when estimating the saving that may be effected by new power. The shop should be regarded as an accessory that is essential to the efficient utilization of the large investment in motive power. Too often the question is decided on the basis of the dircet saving on repair operations, without considering the value of the locomotive-days saved by proper facilities.

14 Wide differences of opinion appear to exist regarding the relative cost of maintenace and mileage of 2–10–2 type and Mallet locomotives. While the field for each is to a certain extent distinct, there are districts where either might be suitable and roads that have sufficient data to permit a fair comparison could perform a service by furnishing information that would clear up this question.

15 A very serious problem in connection with the use of locomotives of high capacity is the effect on the cost of repairs to freight cars. The total amount spent for repairs and renewals is nearly as great as the repairs and renewals to freight and passenger locomotives combined. When the length of trains is increased beyond a certain point, break-in-twos, shifted loads, and damage to the cars in general may increase at a rapid rate. It is not inconceivable that the expense resulting from hidden damage may nullify savings in other items. Local conditions determine whether or not this is an important factor. The effect of increasing the length of the train would be but slight where short, heavy trains of steel cars are hauled. It may be serious where the road cannot control the character of equipment in the trains, where the carload is light, the train long, and the lading is subject to damage or of

such a nature that it may shift and damage the car. It is significant to note that the study of the operation of Consolidation and Mikado locomotives made by N. D. Ballantine showed the time delayed due to car failures was more than twice as great with the Mikado engine (which had a tractive effort of 57,000 lb.) than with the Consolidation of 39,000 lb. tractive effort. No record is available of the cost of repairing the cars involved in these failures, nor the defects noted at terminals that were chargeable to unavoidable shocks incidental to the operation of the longer train.

16 A study of car failures in long trains may demonstrate that the trouble is largely due to equipment with weak underframes. The greater portion of the damage is done to the draft gear and sills, and it is doubtful whether sills meeting the recommendations of the American Railway Association would fail except under the most extreme stresses set up by surging in trains. If wooden underframes are a serious hindrance to the operation of long trains, the remedy can be applied with little difficulty. While the reinforcement of the remaining cars of this type still in service would require fairly heavy expenditures, it would no doubt be justified by the saving in repair costs and improved operation.

TRANSPORTATION EXPENSES

17 The character of the motive power has a decided influence on the expenses falling under this head. Directly or indirectly, the locomotive affects items in the transportation expenses which amount to about 35 per cent of the total operating expenses. However, some of the accounts seem to be affected only slightly, and the importance of locomotives which are merely capable of delivering high tractive effort at low speeds in reducing transportation expenses, has, no doubt, often been overestimated.

18 Two of the important items which are reduced almost proportionately as the tractive effort increases are wages of train enginemen and trainmen. The economies in these expenses are considerable and they can be predetermined with a fair degree of accuracy. Probably for this reason they have assumed undue prominence. In the year 1918 the wages of train employees amounted to 10.5 per cent of all operating expenses. The fuel bill for road engines was practically as much. The cost of locomotive repairs was even greater, and freight-car repairs only slightly less.

19 Some of the costly measures necessary to obtain slight

increases in the train load can probably never be justified on the basis of the savings in the wages of train crews, and some of the related savings are problematical. Railroad officers when considering means of promoting economy might well keep in mind this thought: The gross saving due to a given percentage reduction of the mileage of train enginemen and trainmen under average conditions is equaled by the saving due to a like reduction in the consumption of fuel.

20 Passing to the consideration of the relative fuel consumption of heavy locomotives, each increase in size results in slightly better fuel performance, provided that similar care is used in the design. The essential features of an efficient boiler are large grate area, ample firebox volume, and tubes of suitable length to prevent excessive losses in the waste gases. All these can be obtained in the 2–10–2 type or other heavy locomotives. The large cylinders used with such power are also advantageous because the smaller ratio of the area to the volume reduces the heat loss in the cylinders. Within the range of normal operation, however, the difference in the fuel consumption per unit of work with well-designed locomotives of the 2–10–2 type and of the Mikado type, for example, is probably negligible.

21 The remarkable fuel performance credited to some designs of Mallet compound locomotive suggests the advisability of establishing in as conclusive a manner as possible the comparative results of this type of compound and typical large simple engines.

22 The expenses directly chargeable to train service aside from fuel and wages include lubricants and locomotive and train supplies. These items are of less importance than those previously mentioned, and in general the charges per ton-mile decrease as the tractive power increases.

23 The expense of yard operation is seldom considered as being influenced by the character of the road engines. In hump yards the cost of switching is probably independent of the length of the train. However, in drilling yards the necessity of hauling long cuts of cars reduces the speed of switching and increases the fuel used. While the net result is largely dependent on local conditions, this factor is of some importance and should not be passed over lightly in analyzing problems of operation with heavy power.

24 Enginehouse expenses likewise are affected by the character of the power to an extent depending on local conditions. The reduction in the number of units handled will probably cause a

slight decrease in the cost per ton-mile unless the new equipment makes additional facilities necessary.

25 While the major items of operating expenses which enter into the problem of large locomotives have been discussed above, it is pertinent to enumerate several miscellaneous items which are affected to some degree. These include accounts affected by collisions and derailments loss and damage, damage to live stock, clearing wrecks and injuries to persons. The effect of car failures on loss and damage has already been mentioned. In so far as these expenses are due to collisions and derailments, they are increased by an increase in train density rather than by an increase in the length of the train, and would therefore be reduced by the use of locomotives of high capacity.

FIXED CHARGES

26 The only fixed charges on road and equipment which appear in the operating expenses are the depreciation charges on certain parts of the plant. While the separation of interest charges from labor and material may be desirable in the general balance sheet, the analysis of the advantages of various facilities is best made by considering the net amount that can be earned above the prevailing rate of interest.

27 The fixed charges on the motive power seldom exceed three to four per cent of the operating expenses. The difference between the fixed charges on a thoroughly efficient modern engine and a crude design that might be bought to make an insufficient appropriation cover a given number of locomotives, is negligible. However, the difference in the earning power of these two types is quite appreciable and serves to show what large returns can be derived from the additional capital expended for refinements and accessories that give increased capacity and efficiency.

28 Reference has already been made to the advisability of considering the capital expenditure required for related facilities when deciding on the type of power. At first thought it might seem that the additional investment for terminals, shops, and shop machinery would add greatly to the capital expenditure and the fixed charges. Under ordinary conditions the cost of roundhouse space required properly to house a locomotive is a comparatively small proportion of the cost of the power. The cost of the shop

buildings and machinery is even less important when the added efficiency and decreased cost of repair operations are considered.

29 Sometimes the introduction of heavy locomotives necessitates strengthening or replacing bridges or laying heavier rail over certain sections. The expenditures involved are often quite large, but the relatively long life of these structures decreases the fixed charges and the additional cost per ton-mile becomes comparatively small and is seldom an important factor in determining the most economical equipment. When new rail must be laid the additional expenditure is a more serious item and the charge would probably not be justified except on a line with relatively dense traffic.

30 The preceding discussion pointing out the more important considerations involved in a study of the economic value of various types of motive power demonstrates the complexity of the problem. Probably no absolutely correct analysis is possible; surely, it is not practicable. The question is of extreme importance because the possibilities of economical operation are circumscribed by the motive power. For that reason the choice of the locomotive should be made with extreme care. The final decision should be based on a definite knowledge of the economies that can be realized, not on unsupported opinion. Engineering methods are essential in working out the solution and the study of the problem offers a field for constructive coöperative work by the members of the engineering societies.

DISCUSSION

JOHN E. MUHLFELD. The paper refers more particularly to the question of the class and capacity of steam locomotives which will be best adapted for the various physical, traffic and operating conditions on individual lines in different regional districts, and the writer agrees with the general idea of the author that there are many involved factors to be considered in the determination. At the same time the steam locomotive of great power has so conclusively demonstrated its utility and economy for the movement of heavy tonnage freight during the past twenty years that the writer doubts very much whether any railroad executive would feel disposed at this time to purchase freight locomotives, of practically indefinite useful life, of less capacity than the maximum

allowable for the existing or proposed clearance, track, bridge and train length limitations.

From the writer's experience "the cost per 1000 gross ton-miles moved" is the one that should govern in making decisions on the kind and capacity of new freight locomotives, and this cost should properly cover investment, taxes, upkeep, operation and all other expenses relating thereto. On that basis he does not know of any particular case where a combination of a maximum of tractive power, trainload, gross ton-miles per pound of available locomotive tractive power, revenue load per car, and of car-miles per day, together with the minimum allowable empty car movement, will not produce the greatest net earnings.

To illustrate this point by a practical case, reference is made to the present status of the Kansas City Southern Railway which has, during the past fifteen years, been managed under a constructive and progressive policy. Comparing the annual report for the year 1920 with that for 1910 it will be found that there has been

a A decrease of 8.87 per cent in the total number of locomotives, but an increase of 28.05 per cent in the total tractive power and of 40.5 per cent in the average tractive power of all locomotives

b An increase in the average tractive power of all locomotives from 29,904 to 42,020 lb., and of freight locomotives from 34,628 to 48,400 lb.

c An increase of only 1.63 per cent in total locomotive-miles, but an increase of 20.14 per cent in total freight-car-miles and of 41.22 per cent in freight gross ton-miles

d An increase of 74.4 per cent in the freight-train load

e An increase of 54.6 per cent in the average miles per freight car per day

f A decrease of 1.89 per cent in the empty cars per freight train and of 20.04 per cent in the empty freight-car mileage.

While substantial general improvements in the property, and the effective direction and management of the same have had much to do with this change in performance, the large locomotive has been a predominating factor as referred to in the last part of the first paragraph of the paper.

However, in Par. 15, it is stated that "a very serious problem in connection with the use of locomotives of high capacity is the effect of the cost of repairs to freight cars" and "it is not inconceivable that the expense resulting from hidden damage may nullify savings in other directions." Also in Par. 17, it is said "the importance of locomotives which are merely capable of delivering high tractive effort at low speeds in reducing transportation expenses has, no doubt, often been over-estimated." This, in the writer's opinion, is not so questionable as relating either to the locomotive of great power, the track over which it operates, or to the character of equipment in, or the length of, freight trains, but is more applicable to the manner of locomotive and train handling. For example, in the case of the Kansas City Southern, the 95-mile mountain division over which the first-built 0-6-6-0 type Mallets, of 81,900 lb. tractive power in compound and 98,280 lb. in simple gear, operated were laid with rail as follows:

9	miles	of	75-lb.	rail,	13	years	old
7	"	"	80	"	"	11	"	"
34	"	"	80	"	"	10	"	"
7	"	"	80	"	"	9	"	"
19	"	"	80	"	"	8	"	"
19	"	"	80	"	"	7	"	"

a total of 95 miles.

This rail was laid on unplated oak ties and the track was ballasted with cinders, gravel or rock. The profile is undulating with the majority of the grades 1.35 per cent and a four-mile ruling grade of 1.5 per cent over which latter the Mallets were operated in simple gear. The most of the curves are from 4 to 6 degrees, compensated.

The maximum single Mallet trains hauled during the first three months of service in 1912 averaged from 2130 to 2195 tons in from 43 to 55 cars, northbound, and from 1890 to 2029 tons in from 68 to 79 cars southbound. The freight cars making up these trains were generally of light, wooden construction and principally of 30- and 40-tons marked capacity.

Immediately after these locomotives were put into service the distribution of freight-car drawbars pulled out in through-tonnage trains were recorded and may be compared as follows:

Engineer	Month of Service			
	First		Third	
	Trips run	Drawbars pulled out	Trips run	Drawbars pulled out
A	7	7	21	5
B	6	6	6	1
C	4	4	31	7
D	6	5	—	—
E	6	4	4	1
F	10	3	7	0
G	9	2	—	—
H	13	3	26	1
I	6	1	5	0
J	6	1	15	3
K	—	—	2	4
L	—	—	6	3
M	—	—	25	2
N	—	—	2	1
O	—	—	4	1
P	—	—	14	0
Q	—	—	10	0
R	—	—	10	0
S	—	—	2	0
T	—	—	1	0
Total	73	36	191	29

A further analysis of the performance during the first month of service showed that the principal points on the division where draw-bars were pulled out were as follows:

Location	Number of drawbars pulled out
Mile Post A	11
" " B	7
" " C	7
" " D	4
" " F	4
" " G	3
	Total 36

However, during the third month's service only seven draw-bars were pulled out at all of the locations named, the total of twenty-nine having been distributed over nineteen different locations, four having been pulled out at one point, three at another,

two each at five other points, and one at each of the remaining twelve locations.

Furthermore no difficulty was experienced in moving these Mallet trains over this 95-mile division in from six to eight hours between time of terminal departure and arrival.

The foregoing actual performance is cited to indicate what effect competent supervision and training in locomotive, train and brake handling will have on heavy freight tonnage movement, regardless of locomotive capacity or car equipment and roadway conditions.

The author states that "he has made an attempt to develop a general method of determining the savings that might be expected from the use of high-capacity locomotives, but at every step was so hampered by lack of accurate information that the conclusion was finally reached that *there is no method of solving the problem without extensive research.*" This has not been the writer's experience and he believes it entirely practicable to secure comparable data sufficiently accurate to enable the making of correct decisions. For example, the question of the practical advantage of a certain type of new locomotive as compared with existing power, as regards the cost for fuel and engine and train crews' time only, for operation over a particular district, can be compared with reasonable accuracy in the following manner:

1 Locomotive,	A	B
2 Tonnage, north, actual tons,	3940	3950
3 Tonnage, south, actual tons,	4150	4950
4 Time in motion	7 hr. 40 min.	6 hr. 7 min.
5 Time on road,	9 hr. 40 min.	8 hr. 7 min.
6 Crew's time,	10 hr. 40 min.	9 hr. 7 min.
7 Coal, including standby, for trip, lb.,	33,170	26,670
8 100 gross ton-miles per trip,	3821.75	4161.25
9 Locomotive mileage, average annual,	33,945	33,945
10 100 gross ton-miles, per annum,	1,365,511	1,486,815
11 Helper locomotive fuel per trip, lb.,	5000	none
12 Helper locomotive and train crew's time,	3 hr. 20 min.	none
13 Total cost of fuel per trip, including helper, dollars,	95.43	66.68
14 Cost of lead locomotive and train crew's time, dollars,	63.52	51.62
15 Cost of helper locomotive crew's time, dollars,	8.55	none
16 Total cost of fuel and crew's time, per trip, dollars,	167.50	118.30
17 Average cost of fuel per 100 gross ton-miles, dollars,	0.024970	0.016024
18 Average total cost of fuel and crew's time per 100 gross ton-miles, dollars,	0.043828	0.028429
19 Saving per year on 100 gross ton-mile basis fuel only, dollars		13,301.00
20 Saving per year on 100 gross ton-mile basis, total fuel and crew's time, dollars,		22,290.00

The foregoing can of course be elaborated as much as is desired, and the gross ton-mile basis will give a reliable basis for any and all comparisons.

One of the railroad operating cost items, in which the shipping public is particularly concerned is the "out-of-pocket" or transportation expense, which represents wage and material costs that do not enter into the upkeep or betterment of the property and without a doubt a steam locomotive of the maximum permissible tractive power will produce the most economical result in that respect. In 1914 some data were prepared by one of the larger trunk lines, for one of its divisions, to show the relationship between engine and train crew wages for different weights of tonnage trains as reflected in the cost per gross freight ton-mile and the comparison was about as follows:

Gross tons per freight train, tons	Cost for engine and train crew wages per gross ton-mile, dollars
200	0.00090
300	0.00060
400	0.00045
500	0.00035
600	0.00030
700	0.00025
850	0.00020
1200	0.00015
1700	0.00010
3000	0.00005

Coming now to the advantages of the 2-10-2 type of large freight locomotive as compared with the Mallet type, the writer has taken the Pennsylvania Class 1 I-S Decapod instead of the 2-10-2 as representative of the best features of five pairs of coupled driving wheels, and Table 1 shows this locomotive compared with the original Baltimore and Ohio and various later designs of Mallets for road freight service. In general, the Mallet type, by means of its large percentage of adhesive to total engine weight, and its selective compound or simple operation at starting speeds, has the maximum reserve power for starting heavy trains out of yards and sidings, and the double expansion in the use of steam at running speeds will substantially reduce fuel consump-

tion per unit of work performed. Furthermore, in the Mallet, the distribution of stresses reduces the weights of the various parts that must be handled to maintain wear and tear and the less concentrated driving wheel loads, reduced weight of unsprung reciprocating and revolving parts, and better counterbalancing, in combination with the short rigid and articulated total wheelbase makes it a much easier locomotive on rails, ties, frogs, crossings and bridges which have become a very large item of roadway maintenance.

Therefore, summing the conditions, it is the writer's opinion that the disadvantages of the 2–10–2 and Decapod as compared with the Mallet types of locomotives may be stated as follows:

a Enormous concentrated stresses on cylinders, frames, pistons and rods, crossheads, main rods and main crankpins, main driving axles and boxes, side rods, and main and side rod brasses
b Excessive bridge and rail loads
c Extraordinary tire tread and flange and driving wheel hub and box end-play wear
d Liability of breakage and crushing of parts subject to high concentrated stresses
e Inability properly to counterbalance revolving and reciprocating weights in main driving wheels
f Difficulty in engine house maintenance on account of extreme weight of parts to be handled by mechanics and other labor in making running repairs.

It is frequently brought out that Mallet type locomotives are difficult and expensive to maintain and that special facilities are required for engine house and back-shop repair work. This has not applied on the Kansas City Southern where the large 2–8–8–0 classes, which develop over 103,000 lb. tractive power in compound and 123,000 lb. in simple gear are maintained with the usual engine house and shop equipment, make from 80,000 to 85,000 miles between classified repairs as compared with from 60,000 to 70,000 for Consolidations, and during busy months will all average as high as 3500 miles per locomotive.

With respect to running repair costs which are an index of the cost and time required to keep locomotives available for use, the following comparison will give an idea of how these run.

Kind of locomotive	Tractive power rating, average, lb.	Cost for repairs per mile run, dollars,	Cost per lb. of tractive power mile unit, dollars
Consolidation (2–8–0)	44,800	0.1392	0.0071
Consolidation (2–8–0)	53,850	0.1974	0.0090
Mallet (2–8–8–0)	103,000 (compound)	0.2112	0.0050

TABLE 1—COMPARISON OF DECAPOD AND MALLET LOCOMOTIVES

Railroad	Pennsylvania	Kansas City Southern	Kansas City Southern	Virginian	Baltimore and Ohio
Type,	Decapod	Mallet	Mallet	Mallet	Mallet
Design,	2–10–0	0–6–6–0	2–6–6–0	2–10–10–2	0–6–6–0
Date built,	1918	1912	1918	1920	1904
Tractive power, lb.,	90,000	81,900	103,019	147,200	71,500
		98,280	123,623	176,600	85,800
Boiler pressure, lb.,	250	225	210	215	235
Cylinders, diameter and stroke, in.,	30½ x 32	22 x 35 x 32	26 x 41 x 32	30 x 48 x 32	20 x 32 x 32
Driving wheels, diameter, in.,	62	56	57	56	56
Weight on drivers, lb.,	342,050	352,000	466,000	617,000	334,500
Weight on leading truck, lb.,	29,750	None	29,000	32,000	None
Weight on trailing truck, lb.,	None	None	None	35,000	None
Weight, total engine, lb.,	371,800	352,000	495,000	684,000	334,500
Weight, total engine and tender, lb.,	–	526,000	694,900	898,300	473,800
Factor of adhesion,	3.80	4.27–3.58	4.52–3.77	4.19–3.5	4.67–3.9
Adhesive weight to total engine weight, per cent	92.0	100.0	94.0	90.0	100.0

A. W. Bruce. At present it is practically impossible to compare engine maintenance costs on different roads, largely, the writer believes, because of the system of charges and bookkeeping. Would it not be advisable to adopt, if possible, a common system for purposes of intelligent comparison. This would also include electric installations for the same reason.

Track maintenance is influenced by locomotives through

1 Axle loading
2 Rigid wheelbase
3 Counterbalance (speed).

We cannot do much with axle loading or speed conditions.

We can, however, reduce the rigid wheelbase through the use of lateral boxes on Mallet type engines, and much has been done along this line.

Engine maintenance is increased largely on account of lack of immediate attention, soon resulting in heavy repairs. Given prompt attention, minor repairs can readily be made. No amount of design or material will offset continuous neglect.

The maximum train on most roads is limited by draft gears, etc., of all the cars. Maximum trains can be handled only through the use of maximum equipment. Attention is drawn to special cases of this used on the Norfolk and Western and Virginian Railroads for coal transportation.

Operating economy is largely relative, depending on service involved. Mallet engines are, of course, most economical at low-speed, heavy-drag service. Simple engines increase in economy with reasonable increase of speed. The engine as a whole is economical in accordance with the train performance for which it is designed. Heavy engines are here, of course, to stay. Their economies vary with their maintenance. Train crew expense is very similar for an engine of 40,000 lb. and for one of 140,000 lb. tractive power. Total expenditure depends on road conditions and train moved. Questions might be raised as to what is being done toward the retirement of lighter power, and if there is any regular and systematic method of doing this.

O. C. Cromwell. While we cannot question that the use of locomotives of greater power is desirable, there are conditions in certain parts of the country which set a limit to the sizes of locomotives. The author has spoken of bridges and track: The writer would say that in addition there are tunnels which have to be contended with and which limit the size. The limitations as to size present a complex problem in the design of heavier locomotives. Limited bearing areas, due to restricted widths over cylinders result in increased maintenance cost. The size of boilers is also affected by these tunnel limitations. The use of a more suitable bearing metal, capable of increased service, would be desirable.

If more attention were given to the subject of lubrication, locomotives would stay out of the shops longer between repair periods. In the writer's opinion, lubrication is not as successful today as it has been in the past. The parts subject to lubrication wear more rapidly than formerly. No advance has been made in the lubrication of cylinders, although much discussion has been heard on the subject.

If designers were to visit shops where locomotives were being torn down for repairs, they would learn where the parts wear most rapidly, and better design would result.

The machine work done on locomotives is much inferior to past standards, and results in increased maintenance costs, on account of more rapid wear and waste of power, resulting from increased friction.

PAUL R. DUFFEY. An important feature in connection with the locomotive which is often overlooked is proper design of the tender. Large types of locomotives such as the Mallet, Mikado and Consolidation, do not seem to be provided with tenders of relative capacity economically to serve the class of power, the water and coal space being inadequate to meet the demands, particularly so on locomotives equipped with modern stokers and superheaters.

Experience with a tender having a water carrying capacity of 15,000 gallons, and coal space of 18 tons net, has demonstrated its practicability when used in connection with consolidation locomotives operating in mountain districts and handling 100-car trains. It is found that on a division 89 miles long it is necessary to take water only once and no stop for coal, thus eliminating to a great extent danger of parting train in stopping at and starting from coal and water stations; a saving in time over the division is also made due to minimum number of stops.

The tender referred to is of the low-long cistern type, is constructed on a one-piece cast-steel frame, and has six-wheel trucks, truck frames cast in one section; steel wheels are used. Such a type of tender can be seen in connection with the Western Maryland Railways Consolidation Locomotives, class H-9, numbers 800–840 inclusive.

THE AUTHOR, in general, agrees with Mr. Muhlfeld regarding the desirability of utilizing locomotives of the greatest allowable capacity. However, some of the statements in Muhlfeld's discussion are apparently not well substantiated.

In questioning the effect of locomotives of high tractive effort on the cost of car repairs, Mr. Muhlfeld submits some interesting data. The statistics showing the decreases in the number of drawbars pulled out with Mallet locomotives after the enginemen became familiar with the new power show the importance of careful

handling in reducing this item of expense. However, the cost of replacing draft gear is only a small part of the total cost of car repairs which may be influenced by the size and type of motive power, and, as no figures regarding the total cost are given, the information must be considered as inconclusive.

The tabulation showing the comparative cost of fuel and crews' wages includes only the most easily determined factors involved in the comparison of the cost of operation with two types of power and neglects many of the factors mentioned in the paper.

The table of costs of engine and train-crew wages per gross ton-mile shows strikingly the diminishing value of increasing the trainload beyond a moderate tonnage. For example, the increase of the gross tons per freight train from 700 to 850 reduces the cost $0.00005 per ton-mile, but a change from 1700 tons to 3000 tons, increasing the tonnage 1300 tons or 76 per cent, produces no greater saving, on the ton-mile basis, than an increase of 150 tons, or 21.5 per cent at the lower tonnage. Comparatively slight changes in locomotive or car repair costs, fuel, roadway maintenance, or other items may wipe out the minute savings in train crews' wages effected by a slight increase in the high tonnage where the train load has already been brought to a reasonably high figure.

It would seem that Mr. Muhlfeld has not entirely avoided one of the common fallacies which the author has attempted to point out. The paper was an attempt to show that thorough analysis is needed to prevent the acceptance of the gross tons per train as a measure of the overall efficiency of operation from causing serious losses in expenses where the costs are not easily determined, thus overbalancing the savings in the unit costs of crews' wages and related items.

No. 1796

THE DESIGN OF LARGE LOCOMOTIVES

By M. H. Haig,[1] Topeka, Kan.
Non-Member

After discussing various restrictions and limitations in the design and construction of large locomotives due to the inability of existing bridges and tracks to of bridges and structures, the author expresses the opinion that the physical conditions of a road should be adjusted to the requirements of the locomotive, and that the only controlling factors should be the size of train and the traffic of the territory.

Leading features of locomotive construction such as relative size of cylinders, total heating surface, grate area, principal dimensions, etc., have been treated at length by various writers. Features which have not been so discussed, however, are those which keep a locomotive in service a maximum length of time, reduce engine failures to a minimum, reduce cost of maintenance and repairs, and increase revenue-earning power. And it is in order to arouse interest in those details which are not always given the attention to which they are entitled, that the author proceeds to their consideration. These, in the order of their treatment, are: Counterbalance, cross-heads, driving wheels, crosshead pins, piston rods, cylinders, frame braces, boiler cracks, back flue sheet, grate rigging, water columns, cab equipment, cab seats, and tender capacity. The author's opinions on these topics are presented in the hope that discussion will bring out the experience of others not only on these but on additional details of locomotive design and construction.

THE design of a large locomotive depends on the service to which it is to be assigned. The service varies with the weight of the train to be hauled and the number of cars in the train, and is affected by the topography of the territory on which it is to operate, ruling grades in each direction, length of grades, average speed between terminals, method of dispatching, whether single or double track between terminals, etc. This information being available, it is a reasonably simple matter to determine upon the leading features of a locomotive to meet the requirements.

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RESTRICTIONS AND LIMITATIONS IMPOSED

2 For a locomotive to give practically 100 per cent service, its design and construction must not be restricted by personal opinion or by physical limitations of the road. If the weight needed for adhesion in starting a given train is restricted by an opinion that certain wheel loads should not be exceeded or because bridges and track are not capable of carrying the necessary weight, then the capacity of the locomotive is restricted and the train must be adjusted to the locomotive, instead of the locomotive being built to suit the train. This in turn has a tendency to limit a division or a railroad as a whole. Limitations such as these, together with clearances of bridges and structures, obstructions along the right of way, etc., affect locomotive design and construction. The locomotive as a whole is dwarfed, or some of its vital and essential parts are so dwarfed as to cripple the machine as a whole.

3 A railroad is a plant, establishment or organization for manufacturing transportation. The locomotive is a very important part of the plant and is one of the most direct earners of revenue from which the transportation-manufacturing plant obtains its income. As such, it is a matter of business and economic principle to adjust some of the physical conditions of the road to meet the requirements of the locomotive, to prevent dwarfing it and to prevent sacrificing its power. Meeting these requirements of the locomotive amounts to meeting the necessary requirements of traffic. No turntable installed at a principal roundhouse should be less than 100 ft. long, and in many cases the length should be 125 ft. The distance between walls of a modern roundhouse should be great enough to permit closing the door behind the tender of a Sante Fe or Mikado type locomotive and have ample room for trucking between the locomotive pilot and the outer wall of the roundhouse. Passing tracks should be long enough to take trains justified by the business and traffic of the division or territory. Bridges, rail and roadbed should be capable of carrying a static wheel load of at least 65,000 lb. per pair and of permitting the additional stresses resulting from a freight speed of at least 45 miles per hour. In meeting the requirements of rail stresses particular attention should be given to the employment of heavy rails on curves.

4 Unless these physical conditions are provided, a locomotive cannot be designed and constructed without restriction and proper power cannot be furnished to meet requirements. The only governing factors should be the size of train and the traffic of the territory.

LEADING FEATURES OF LOCOMOTIVE CONSTRUCTION

5 Leading features of locomotive construction, such as relative size of cylinders, length of stroke, total heating surface, superheating surface, grate area, etc., have been well covered by handbooks and pamphlets issued by locomotive builders and by reports to the various associations, as well as by articles in the technical press. Tables of principal dimensions of large locomotives are obtainable from the same sources, together with detailed descriptions of features of design and construction which have met with general favor and some which have been short-lived. A discussion or comment on these features would therefore be largely a repetition of facts already presented and easily available.

6 Features which have not been so generally discussed and exploited are those which keep a locomotive in service a maximum length of time, reduce engine failures to a minimum, reduce cost of maintenance and repairs, and increase revenue-earning power. Among these, durability of material and accessibility of parts are important factors. The latter implies arrangements by which a locomotive is made free from complications in construction, inexpensive to repair, easy to maintain, and so put together that needed repairs can be made handily and quickly.

7 Almost as important as providing a locomotive that will meet the requirements of trains to be hauled and traffic conditions, is providing one that requires minimum repairs — a locomotive that after one trip is ready to be turned for the next trip.

8 A locomotive is in revenue-earning service only when it is hauling trains. Any road can make a study and determine what proportion of its locomotives are unserviceable and what percentage of the time its serviceable locomotives are on the road. Such information will show what percentage of the time its engines are earning revenue.

NECESSITY OF COÖPERATION IN LOCOMOTIVE DEVELOPMENT

9 There are a number of factors which determine the percentage of time a locomotive is in service. While it is beyond the scope of this paper to attempt to discuss all of these, it may nevertheless be noted that durability of material is a vital one.

10 To maintain the advantages of designs already existing and to develop these still further requires the unlimited coöperation not

only of the mechanical, civil-engineering and operating forces of the railroads, but also of the locomotive builders and particularly of the manufacturers of material.

11 On an individual railroad there must be a well-developed system by which the forces responsible for design are kept informed of conditions under which its locomotives operate — of conditions common to the whole system and of conditions peculiar to local territories. They must be informed of the shop and terminal facilities and of the physical conditions of the road affecting size of trains, clearances, etc. It is of prime importance that they be advised fully of all failures of parts and that particulars thereof be furnished with all possible correctness. Reports should include causes as far as possible and information by which it can be determined whether failure resulted from weakness in the part or was brought about by some outside cause. A practice by which broken parts are inspected and investigated by the designing force is worth the comparatively small expense incurred by having its representatives visit the various shops and roundhouses and by having the most important broken parts sent to one central point for investigation.

12 The necessity of unlimited coöperation by manufacturers of material is evident from the study of failures of parts both large and small. On the principle of encouraging further consideration of such coöperation by all concerned and for the purpose of arousing interest in those details of locomotive construction which are not always given the attention to which they are entitled, a number of details which seldom appear among "leading dimensions" will be discussed.

COUNTERBALANCE

13 Important among such details and one which is affected particularly by designers and manufacturers of material, is the counterbalance. The blow from the counterbalance is caused by the difference between the weight of the revolving parts carried by the pins and the total weight in the wheel to balance both the revolving and reciprocating weights. In other words, it is the weight in the wheel to balance reciprocating weight that causes the hammer blow.

14 Weight of reciprocating parts therefore affects hammer blow of driving wheels, riding qualities of locomotive, possible damage to track and bridges and total weight of locomotive. It is

particularly essential to make these parts as light as possible, and to make them light. the material must be durable.

15 Due to the increase in weight of locomotives and to the hammer blow on rails when reciprocating parts are heavy, the 1915 Committee of the American Railway Master Mechanics' Association made the following recommendation:

An attempt to counterbalance large locomotives in both freight and passenger service according to this recommendation has demonstrated its merit, but has further demonstrated that the durability

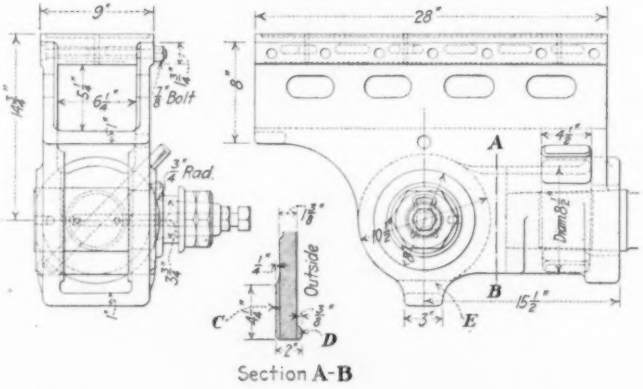


FIG. 1 LAIRD CROSSHEAD USED ON VARIOUS TYPES OF LARGE LOCOMOTIVES

Keep total weight of reciprocating parts on each side of locomotive below $\frac{1}{16}$ part of total weight of locomotive in working order and then balance $\frac{1}{2}$ weight of reciprocating parts.

of both cast and forged steel must be improved if the method is to be continued.

CROSSHEADS

16 The Laird type of crosshead is lighter than several other designs, its performance is very satisfactory in service, and it therefore has advantages in designing for light reciprocating parts. A crosshead of this type used interchangeably on large freight and passenger locomotives is shown in Fig. 1.

17 The construction originally employed is shown by the figure, with the exception of later reinforcement at *C*, *D* and *E*. After about a year's service these crossheads began to break, the

weakness appearing in the relatively thin wall between the hub around the piston rod and the lighter hub around the crosshead pin. The same weakness developed in crossheads of similar general design among locomotives of three or four different classes. The defects which proved common to these different crossheads are shown in Fig. 2.

18 By breaking up these crossheads in order to investigate the

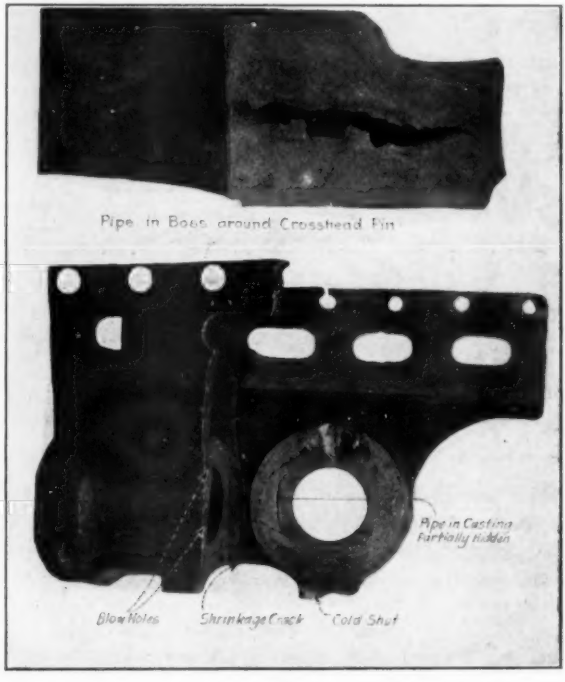


FIG. 2 LOCATION AND NATURE OF DEFECTS IN A POORLY CAST CROSSHEAD

nature of the metal, it was found that in most cases each fracture had its origin in a shrinkage crack. The metal in most of the broken crossheads was found to be porous and to contain blowholes or gas holes, or shrinkage cracks, cold shuts or pipes. In some cases all of these defects were present.

19 The cylinder of the locomotive using the crosshead shown in Fig. 1 is 27 in. in diameter and the boiler pressure 190 lb. The stress in the cross-sections of the two walls at AB (considering only

an area above center line of piston rod equal to area below center line of rod) is 4660 lb. per sq. in. for the original crosshead and 3940 lb. per sq. in. for the reinforced crosshead. Considering the sectional area of the walls from top to bottom, the stresses are respectively 2500 lb. per sq. in. for the original crosshead and 2430 lb. per sq. in. for the reinforced crosshead. According to the average engineering handbook, the ultimate strength of cast steel is 70,000 lb. per sq. in., and with a factor of safety of 10 a safe limit would be 7000 lb.

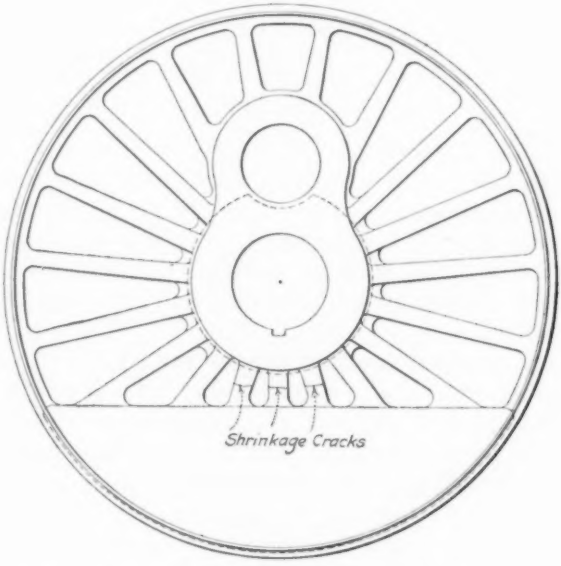


FIG. 3 CRACKS IN DRIVING-WHEEL SPOKES

The stresses above quoted are well within this limit and provide additional strength for stresses of inertia and other causes.

20 Fig. 2 shows very clearly the difference in cross-section of the metal at and near the break. This difference is no doubt largely responsible for the defects in the metal which have caused an epidemic of failures. Crossheads of this general design have been used for many years, and as it appears impossible to modify the shape to advantage, the question, then, is whether foundries can adjust their practices to cast such irregular sections without blowholes, shrinkage cracks and other defects. This is one of the op-

portunities for manufacturers of material to coöperate with the locomotive designer.

DRIVING WHEELS

21 Another irregular section which causes shrinkage cracks is the cast-steel driving-wheel center. Rims and spokes are of much lighter section than the hub and counterbalance, and shrinkage cracks are not unusual at the juncture of these light and heavy sections. An example of this is shown in Fig. 3. Foundries which cast locomotive parts have these conditions to meet and it is believed that foundry practices can be adjusted to meet them.

CROSSHEAD PINS

22 In the development of locomotive construction within recent years the union link of outside valve gears has been connected direct to the crosshead pin. This reduces weight by eliminating the crosshead arm and by shortening the length of the combination lever, thus lessening reciprocating as well as total weight. A further advantage is in eliminating the bolted connection between the crosshead and arm.

23 In eliminating the crosshead arm the duty of the crosshead pin is increased. A broken crosshead pin is more serious than a broken crosshead arm. When a pin breaks there is possibility of something else being broken and a very great probability of a cylinder head being knocked out and carrying a part of the cylinder wall with it.

24 It is therefore absolutely necessary that the material in the crosshead pin shall be of a good quality. This is illustrated by the case of twenty-five large locomotives for which the crosshead pins were furnished from two different sources. The pins from one source on ten of these locomotives gave no trouble. The pins on the other fifteen locomotives were from another source and later development proved that they were made of a very poor quality of steel. The pins of the first ten were made with a good fillet at the inner end of the union link shank and the pins of the second fifteen without fillets. Among the fifteen, several failures occurred which damaged cylinders, and to prevent further failures all of these pins were removed and replaced by new pins of better steel. Analysis of the steel that failed showed it to contain 0.11 per cent carbon, and under test its tensile strength proved to be less than 50,000 lb. per sq. in. Pins

for this service should be made from steel containing about 0.50 per cent carbon and having a tensile strength of 80,000 lb. per sq. in.

25 By reference to Fig. 1 it will be observed that the diameter of the union link shank of the crosshead pin is $3\frac{3}{4}$ in. This is believed to be considerably larger than usual in locomotive design. Even though the stresses in the crosshead pin are low, this large size appears to be a necessary precaution against the uncertainty in quality of material. As a further precaution there is a $\frac{3}{4}$-in. fillet at the end of the shank.

PISTON RODS

26 The most frequent locations of breaks in piston rods of at least one railroad are shown in Fig. 4. The greater number of breaks have been through the keyway. Next in order is the location in the crosshead fit adjacent to the collar. Breaks in the body are



FIG. 4 LOCATION OF BREAKS IN PISTON RODS, NUMBERED IN ORDER OF THEIR MOST FREQUENT OCCURRENCE

usually adjacent to the collar at the crosshead fit and occasionally at the collar adjacent to the piston-head fit.

27 The mechanical fit between the rod and the crosshead is often responsible for the breakage of the former. If there is not a good bearing throughout the length of the fit or at both ends of it, there is opportunity for a slight movement of the rod in the crosshead. This starts a crack in the rod on the same principle as breaking a piece of wire by continuing to bend it at one point, and the crack gradually progresses into a fracture. To facilitate making a good bearing at both ends of a piston-rod fit in a crosshead, the diameter is reduced $\frac{1}{16}$ in. for a length a little greater than the keyway and about midway between ends of the fit. See Fig. 5. To prevent cracks starting in sharp corners at edges of the keyway, these edges are chamfered as shown, the chamfer being made at both ends of the keyway and entirely around.

28 Rods with comparatively low stresses sometimes fail in such a manner that it is difficult to attribute a cause unless quality of material is responsible. This is true of rods of ordinary carbon

steel as well as of those of specially refined steel and of alloy steels. It is more frequently than otherwise that a rod cracks halfway or more across before failure occurs. If half the area or less is sufficient to meet the stresses before a progressive fracture finally occurs, it would seem that the excess area employed is the result of uncertainty regarding the quality of the metal. In this is another opportunity for the assistance and coöperation of manufacturers of material.

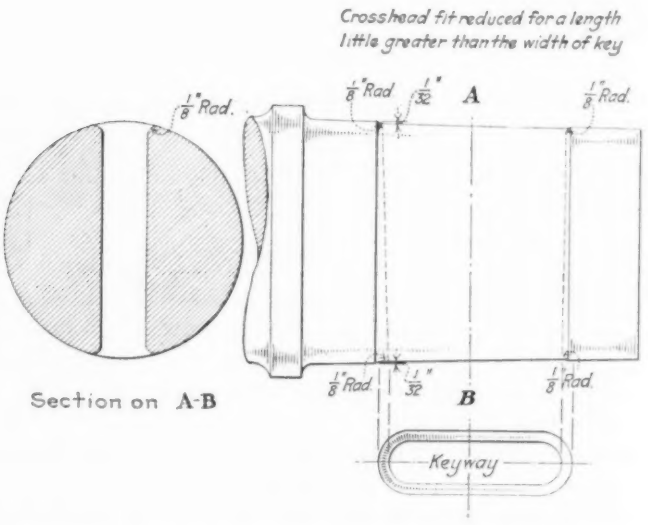


FIG. 5 PISTON-ROD FIT IN CROSSHEAD WITH MIDDLE OF PIT RELIEVED AND KEYWAY EDGES CHAMFERED

CYLINDERS

29 Failures of parts such as those described in preceding paragraphs, quality of material, uncertainty of cylinder cocks being operated, extreme variation in temperature due to use of superheated steam, foundry practices, etc., all affect the design of cylinders. Consideration of these and other features has resulted in the development of the design shown in Fig. 6, which is that of the cylinder of a Mikado locomotive. Except for modifications in dimensions, this represents cylinders used also on Sante Fe, Mountain, Pacific and other locomotive types. The principal features in this cylinder are:

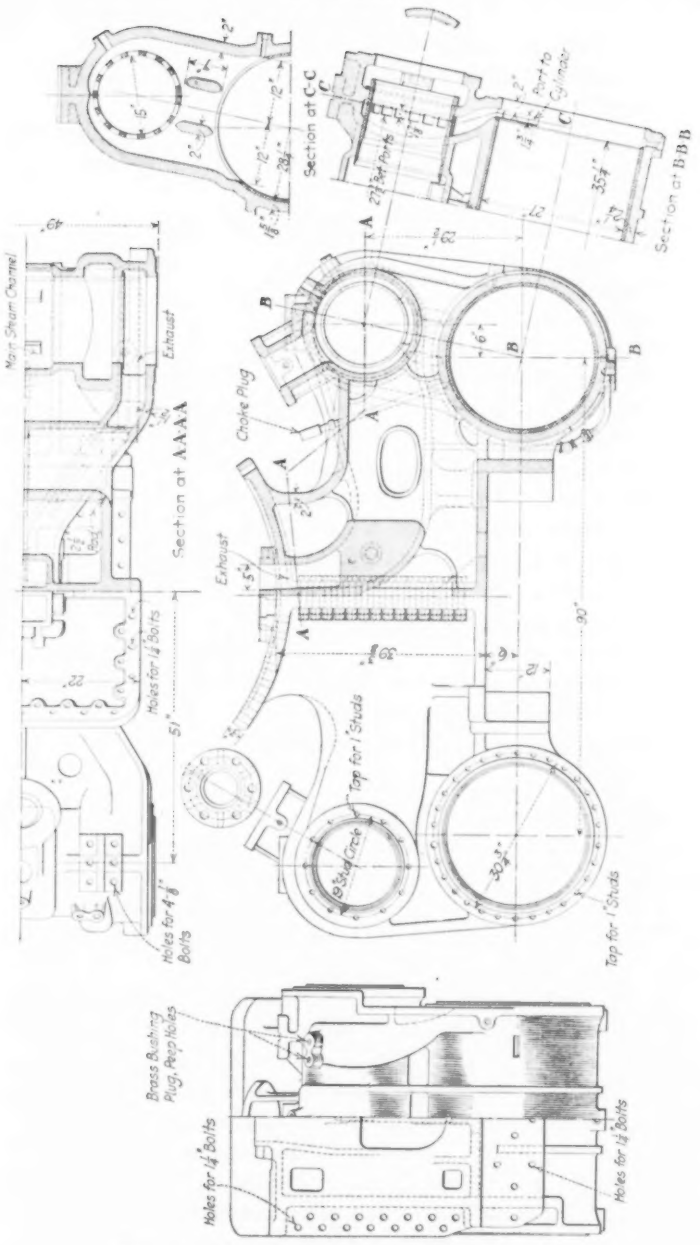


FIG. 6. CYLINDER FOR LARGE LOCOMOTIVE

Simplicity in construction
Uniform thickness of metal
Absence of heavy metal sections at junctions of walls
Walls and parts of ample thickness for strength, well ribbed, well braced and arranged with easy curves and generous fillets
Uniform sectional area throughout length of steam and exhaust passages
Short steam ports
Small steam clearance
Sections of metal, fillets and other features arranged to eliminate internal stresses set up in metal when cooling
Double row of splice bolts holding halves of cylinder saddle together
Double row of bolts at smoke arch
Triple row of horizontal bolts securing cylinder casting to frame
Depth of saddle casting directly above frame forming a box section and providing strength where shallow castings used with double-frame rail failed in the past
Location of valve close to cylinder, permitting short ports, straight valve gear without offsets, and application of nearly straight steam pipe
Large openings to cylinder cocks.

The steam and exhaust channels are free from obstructions and restrictions which will interfere with free flow of steam. The exhaust channels are gradually reduced in area from valve bushing to base of exhaust pipe in such a manner that the cross-sectional area of any point in the channel is not larger than any area through which exhaust steam has previously passed.

30 A weakness in castings of some large cylinders has been in the wall around the live-steam port. As shown in Fig. 6, this wall is made 2 in. thick and the distance across the port below the valve bushing is 24 in. To reduce stresses in this wall it has been made thicker than most other walls of the casting, and, compared with former practice, width across port has been reduced about 4 in. The bridges in the live-steam port are 2 in. thick. They were formerly but 1 in. in thickness and it was not unusual to find them cracked clear through. The change was made to increase the cross-section

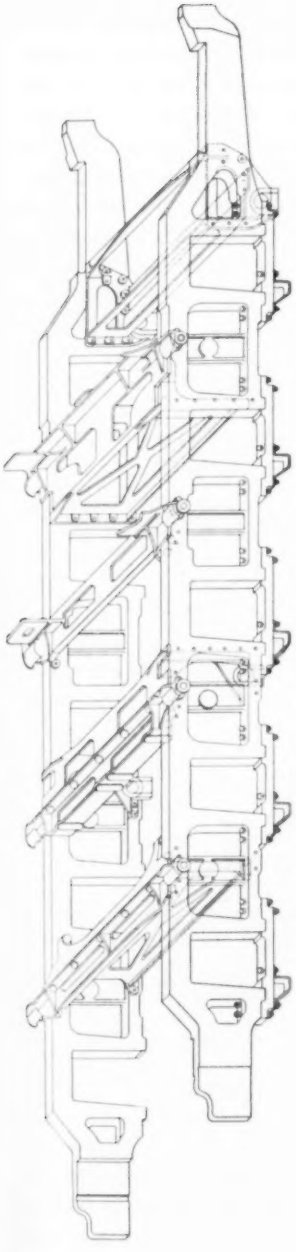


FIG. 7 FRAMES AND FRAME BRACING OF LARGE LOCOMOTIVES

of the bridges in relation to the adjacent walls and thereby reduce tendency to shrinkage cracks.

31 To obtain a good cylinder casting from any design, it is necessary to have proper coöperation of the pattern shop and the foundry. Patterns must be well built and carefully checked. The checker should exercise especial care to see that pattern-makers apply all the fillets called for. The foundry should so arrange the molds as to obtain uniform sections of metal. To insure this, careful measurements should be taken when cores are being set and a drop light should be let down into the mold when taking measurements.

FRAME BRACES

32 Locomotive frames are subjected to repeated lateral and twisting stresses, as well as to various other stresses, which will gradually break a single frame, but which can be withstood indefinitely by the application of substantial braces. An example of a pair of strong frames substantially braced is shown in Fig. 7, the arrangement illustrated being for a Santa Fe type locomotive. Bracing for a Mikado type locomotive is practically the same except that there is one less brace. Bracing in the manner shown has been used for a number of years very successfully, and with very little modification is applicable to any locomotive class with outside valve gear.

33 It must not be thought that breakages of old frames originally unbraced can be stopped by the application of braces. Frames must be made large enough around bolt poles so that frames will not be weakened when bolt holes are drilled. Also, the braces must be applied before the frames have become crystallized and before progressive cracks have started.

34 Braces must be bolted to frames securely. Where braces or castings of other parts are bolted to a frame, the bolts should be applied with the head end bearing in these parts and not the thread end. This will provide for bearing on the bolt through the full thickness of the parts bolted to the frame.

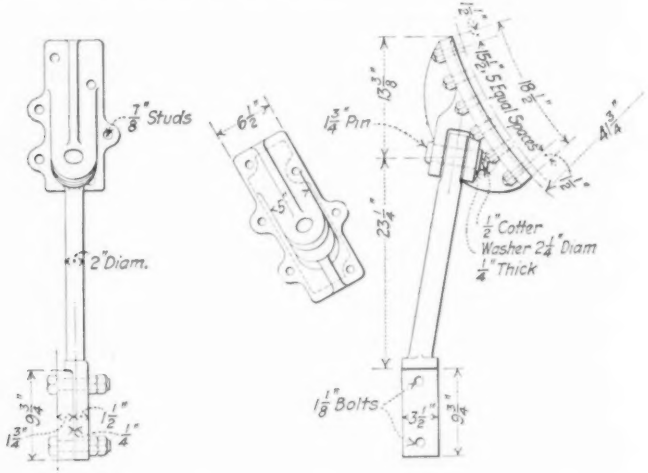


FIG. 8 FLEXIBLE BRACE AND REINFORCING PLATE ON BOILER

35 It will be observed that the vertical strut between the top and bottom frame rails is cast as a part of the frame. Breakages in vertical struts and an investigation into the movement of frames have demonstrated that strength in these struts is necessary. The form of strut shown requires a double equalizer, but the advantage resulting to the frame justifies the modification in the equalizer.

BOILER CRACKS

36 In using the boiler to supplement the frames in forming a backbone or foundation from which to brace machinery parts, the boiler shell is subjected to additional stresses which result in cracks

in the sheets. These cracks usually start in stud holes, sometimes extending from one stud hole to another and sometimes developing where there are no holes.

37 The most frequent causes of these cracks are guide-yoke braces, valve-motion braces and the ordinary belly braces to frames. Guide-yoke and valve-motion braces are often very stiff and are bolted securely to the frames and studded to the boiler. When the boiler expands, the braces and connections are held rigidly by the

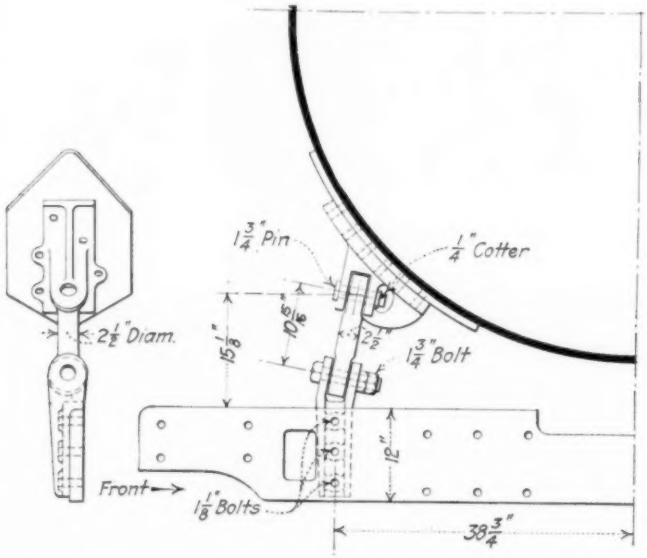


FIG. 9 BOILER BRACE WITH FLEXIBLE CONNECTION AT EACH END

frames and there is a tendency for the boiler to tear itself loose from these fastenings. This sets up strains in the metal which are aggravated by the vibration and pounding to which braces are subjected.

38 In an effort to overcome these cracks, outside welt plates have been riveted to the boiler to reinforce it where the brace pads are studded on. Experiments have been made with flexible, or partially flexible, braces, some of which have so far been successful. An arrangement of reinforcing sheet on boiler and partially flexible brace is shown in Fig. 8.

39 On Pacific type locomotives, for example, with rigid braces, cracks have developed in boilers in the rather short period of

three years. On a number of Sante Fe type locomotives, braces similar to Fig. 8 have broken at the lower end where offset to form a bearing on the valve-motion bearer. These are 11 ft. 7 in. back of the cylinders, and within this length there is room for much expansion in the boiler. Even though there is some expense in renew-

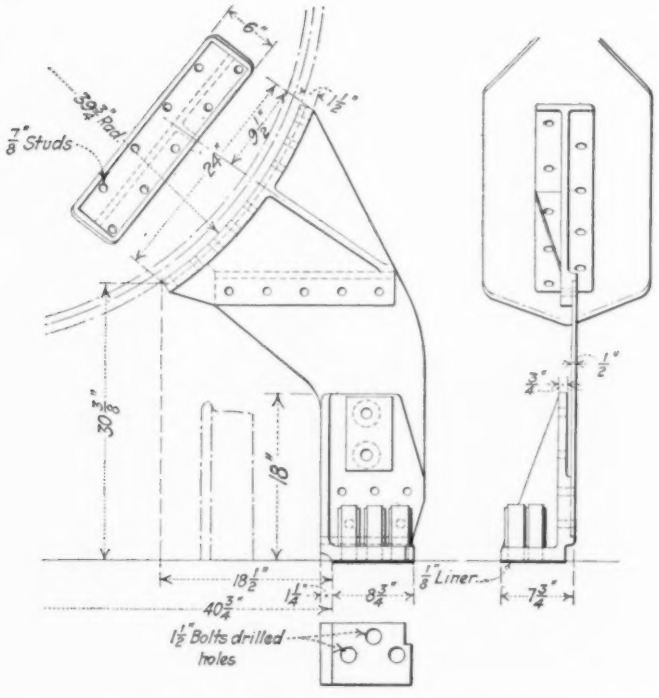


FIG. 10 CAST-STEEL BRACE WITH THIN PLATE TO PROVIDE FLEXIBILITY

ing these braces, it is better to do so than strip the boiler and apply a patch.

40 On engines on which this breakage has occurred, some of the braces are being replaced by braces with a pin connection at the lower as well as upper end, as shown in Fig. 9. Where the use of pins is not favored, however, a thin plate in connection with a cast-steel brace should provide sufficient flexibility for expansion of the boiler and proper stiffness for bracing machinery parts. A brace on this principle is shown in Fig. 10.

THE BACK FLUE SHEET

41 Boiler back flue sheets of large locomotives are renewed and patched more frequently on account of cracks in the knuckle near the top flange than from any other cause. On at least one road the average life of flue-sheet knuckles is 3 years and 3 months, the

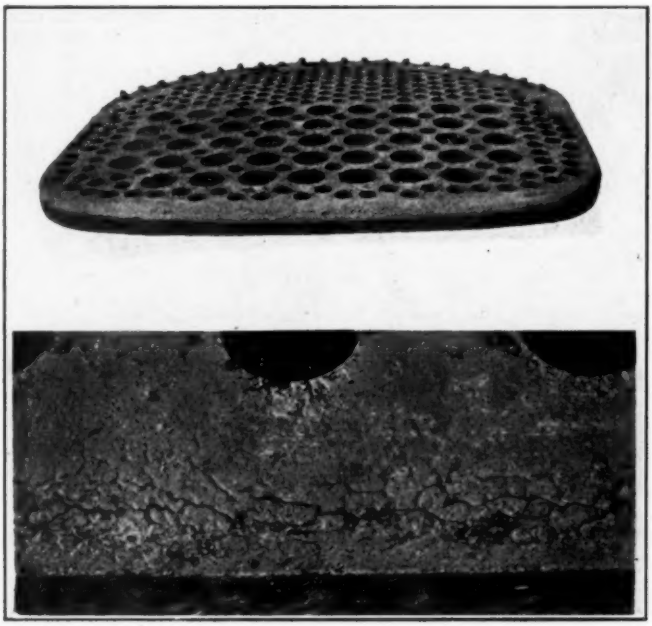


FIG. 11 CRACKS DEVELOPED IN FLUE-SHEET KNUCKLE OF LARGE LOCOMOTIVE

maximum and minimum varying within rather a large range. The opinion is frequently advanced that the cracking is due to the proximity of the flue holes to the top flange. The fact is that conditions which cause cracks exist whether holes are present or not. It is also true that vertical cracks which start in the flue-sheet knuckle tend to progress toward a hole and that minor cracks start in hole and progress upward. The condition of a flue-sheet knuckle after 2 years and 4 months' service is shown in Fig. 11. The top of the flue sheet was cut out and a patch applied because of cracks at the extreme upper part of knuckle and some distance from the flue

holes. By examining the illustration it will be seen that the slight cracks at the edge of the flue hole are undeveloped and are not deep.

42 A minimum limit of distance of top flue holes from top of flue sheet that is considered practical is shown in Fig. 12. To omit flues near the top of the flue sheet sacrifices heating surface. To raise the top of the flue sheet above the usual location of flues increases the weight in the firebox, adds to the amount of water necessary to cover the crown sheet, and by requiring increase in diameter

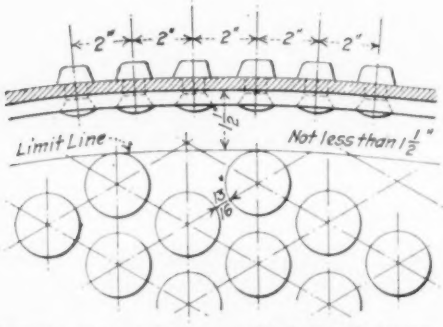


FIG. 12 MINIMUM DISTANCE BETWEEN TOP ROW OF FLUES AND FLANGE OF BACK FLUE SHEET

of boiler to maintain steam space above the crown sheet, increases the weight of the boiler and consequently the weight of the locomotive as a whole.

43 Considering the stresses and the peculiar punishment to which flue-sheet knuckes are subjected, it is important to specify this material carefully and to adhere to the limitations of the specifications. Among details of this specification, the following limits have been demonstrated by experience as practical:

Tensile strength	52,000 to 62,000 lb. per sq. in.
Elongation	Not less than 25 per cent
Carbon	0.12 to 0.25 per cent
Sulphur	Not over 0.025 per cent

THE ASHPAN

44 The ashpan is usally among the last details of a locomotive to be designed. Its importance and usefulness, however, justify its being among the first. Various details at the rear of a locomotive should be arranged to permit a large ashpan with smooth slope sheets at an angle that will permit cinders to fall to the hopper without obstruction. Before deciding on the depth of the firebox throat, arrangement of mud ring, construction of rear section of frame, etc.,

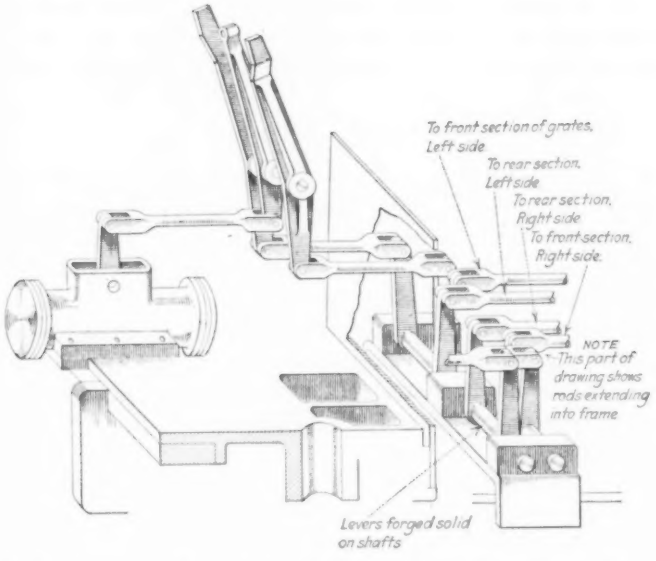


FIG. 13 INTERMEDIATE ROCKERS FOR LOCATING GRATE RODS UNDER CENTER OF GRATES ON LOCOMOTIVE EQUIPPED WITH STOKERS

the ashpan should be given consideration, and the design of a satisfactory pan should be decided on before the designs of surrounding parts have progressed too far. It happens too frequently that locomotives are designed to care for other features and an inadequate, unsatisfactory ashpan is crowded into a limited space.

45 Equally as important is area between the ashpan and mud ring or through parts of the pan, to admit air to support combustion. This area should be at least equal to the area through the boiler flues, and preferably a little greater.

THE GRATE RIGGING

46 The place for grate rods, which operate the grates, is near the center of the grates and above the deep portion of the ashpan. On locomotives without stokers this arrangement is not difficult to provide for. With some stokers, however, grate rods in this position are interfered with, and this has resulted in some grate rods being located along the sides of pans, in certain cases very close to the flat portion or shelf of the pan under the mud ring. In this position the rods collect cinders close to the air openings and obstruct the admission of air for combustion. With steam grate-shaker equipment and stoker the grate rods can be located near the center of grates by applying a set of intermediate rockers as shown in Fig. 13.

WATER COLUMNS

47 In some preliminary tests to determine the best proportions of a water column and connections, use was made of a column $2\frac{1}{2}$ in. in inside diameter provided with a top pipe $1\frac{1}{2}$ in. in outside diameter ($1\frac{1}{4}$in. inside diameter). Experience with the parts of these sizes demonstrated that greater volume is required in the column and greater area through the top pipe. Accordingly a column was made $3\frac{1}{2}$ in. in inside diameter, using a top pipe $2\frac{1}{4}$ in. in outside diameter (2 in. inside diameter), with which arrangement it was found under ordinary circumstances that register of water and gage cocks agreed.

48 Tests were also made to determine the best size of connection to use between the bottom end of the column and the boiler. These demonstrated that a very large opening was undesirable, that too small an opening had a tendency to make the movement of water in glass sluggish, and that a connection having a $\frac{3}{4}$-in. opening was the most suitable.

49 A very thorough investigation into conditions affecting the performance of water columns indicates that the most satisfactory service is obtained with a column and connections conforming with the following specifications:

Inside diameter of water column	$3\frac{1}{2}$ in.
Inside diameter of top steam pipe	2 in.
Inside diameter of connection of column to top steam pipe	Not less than 2 in.
Inside diameter of bottom connection to boiler	$\frac{3}{4}$ in.

Top steam pipe as short as possible consistent with required location forward of boiler back head flange.

Minimum number of bends in top steam pipe to column. This pipe to be lagged.

No valves between water column and boiler either in top steam pipe or in bottom connection.

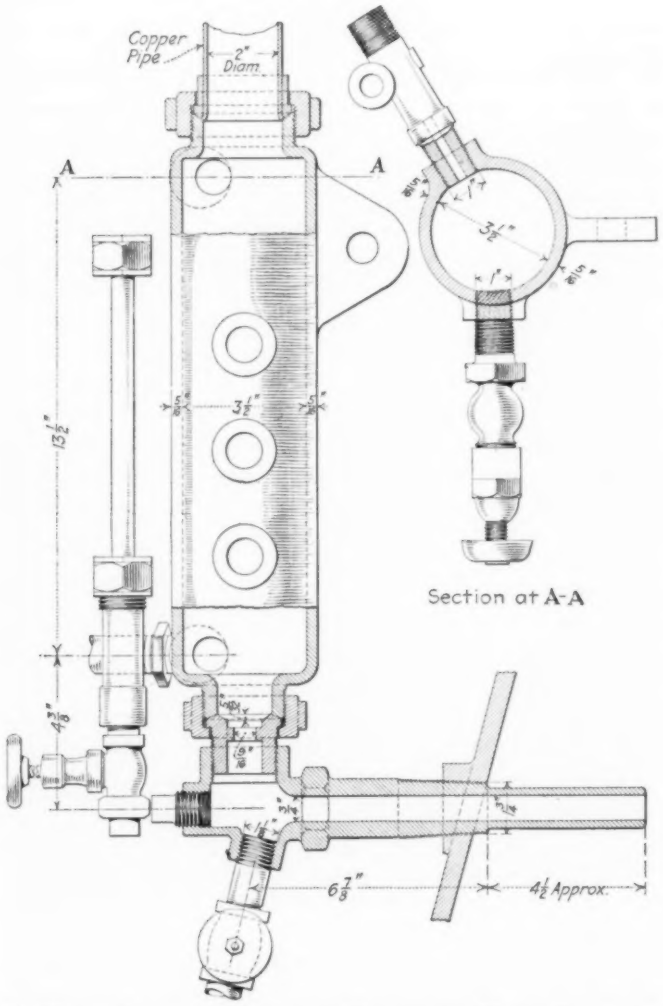


FIG. 14 WATER COLUMN OF SUFFICIENT VOLUME FOR WATER GLASS AND GAGE COCKS TO REGISTER ALIKE

Water-column bottom connection should extend into boiler far enough to clear nearby T-irons or other obstructions, approximately 4½ in. from inside of sheet.

A water column of these principal dimensions is shown in Fig. 14.

CAB EQUIPMENT

50 The satisfactory design of a locomotive is sometimes judged by the extent to which the construction appeals to the convenience and tastes of the individual who is judging. Principal among the features that effect the judgment of enginemen is convenience of cab equipment.

51 The back wall of the cab should be far enough away from the boiler back head to give room for a satisfactory seat, for the application of the required equipment, and for a man to pull the throttle open without striking his arm against it. A distance of 46 in. from the face of back head at the center of fire door to the back wall of cab will meet these requirements.

52 Engineers' and firemen's seats should be located where the men can see ahead and their vision should not be obstructed by air pumps located too high, classification lamps misplaced, running boards too high at the front, or other obstructions that might interfere with their seeing semaphores, switch stands, etc.

53 Blow-off cock handles should be so located that they can be operated by a man in a position where he can see the water glass, and preferably without leaving his seat. The water glass, steam gage, air gages, etc., should be so located that they can be seen by the engineer when in usual position on his seat.

54 The throttle lever, power reverse lever, cylinder-cock lever, sander valves, brake valves, etc., should be located where the engineer can reach them handily when sitting in usual position on his seat or sitting with his head out of the window. It appears like a small detail, but it is a worth-while one to locate the straight air valve where it can be reached easily by an engineer when in such a position that he can see a man at the back of the tank giving signals for coupling to a train.

55 The lubricator must be at such a height that a man can see the feeds, and it must be high enough to avoid pockets in the oil pipes. It must be far enough below the cab roof to be filled easily.

56 The cab equipment requires careful study and it is difficult to locate the various appliances by drawing, but it has been done.

A cab with a large amount of equipment on the boiler back head, yet which is regarded as being reasonably convenient, is shown in Fig. 15.

THE CAB SEAT

57 The use of clear-vision windows has made it somewhat difficult to arrange the seats so that either seat or window will be

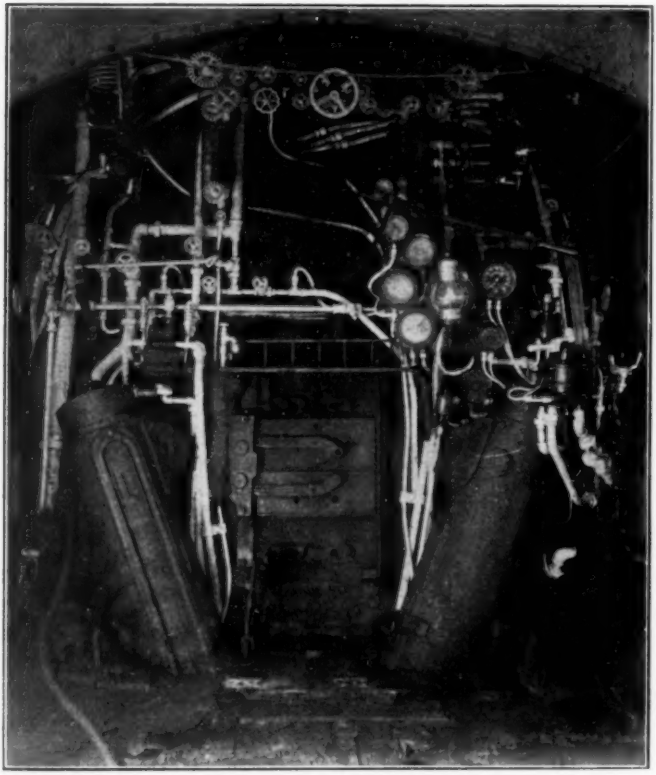


FIG. 15 LOCATION OF BOILER BACK HEAD FIXTURES IN LARGE LOCOMOTIVE

at the height to suit different men. This problem, however, has been solved for one road by its motive-power department chief, who has developed the adjustable seat for engineers and firemen shown in Fig. 16. After continued service of about two years these seats

have been found very satisfactory. They are made of steel, a spring cushion forms the seat, and the back is upholstered. The back being secured to the seat and independent of the back of the cab, prevents any vibrations resulting from shaking of the cab wall.

TENDER CAPACITY

58 Tender capacity should be arranged so as to reduce to a minimum the time a locomotive is detained from the productive work

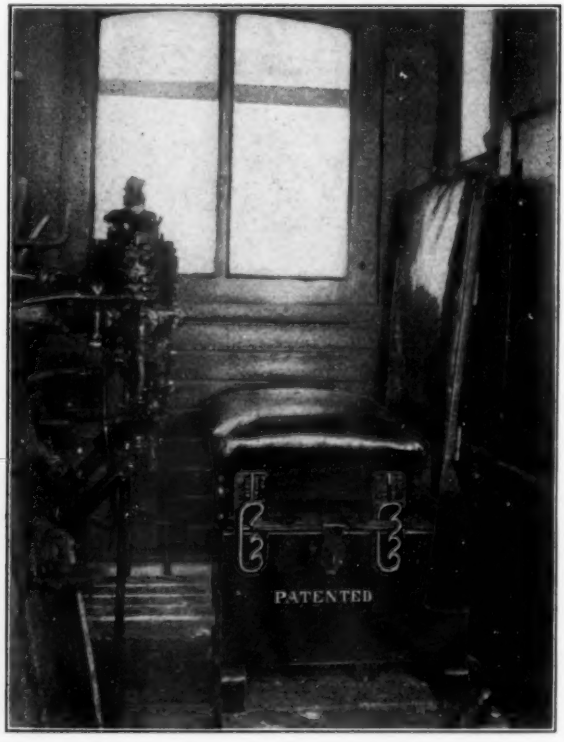


FIG. 16 ADJUSTABLE SEAT FOR LOCOMOTIVE CAB

of hauling trains for the purpose of taking water and fuel. This implies large fuel and water capacities, but in arranging for suitable tender capacity care must be taken to avoid unnecessary weight, as any increase in the weight of the tender produces an equal decrease in the weight of train that can be hauled behind the tender.

59 Tender fuel space should be arranged so as to enable the locomotive to handle a full train with as few stops for fuel as may be feasible.

60 On territories equipped for water to be taken on the run or when stops for purposes other than taking fuel or water are made regularly at stations where water may be taken, the water capacity should be only sufficient to supply the locomotive when handling a full train, between water stations, with a moderate surplus for unusual delays.

61 On territories handling a large percentage of through trains with few stops, tenders of large capacity are desirable as they permit keeping locomotives more continuously at work. Where water is scarce and the supply has to be hauled to water tanks, tenders of large capacity are desirable as they reduce the number of water stations that must be maintained as well as the number of locomotives, cars and men employed in hauling and handling water at these stations.

62 On a certain district 130 miles in length, the substitution of new locomotives having 15,000-gal. tenders for less modern locomotives having 9000-gal. tenders made it possible to discontinue, except for emergency use, two water stations to which water had to be hauled. This resulted in dispensing with two pumpers and in a substantial reduction in the number of tank cars required for distributing water over the territory.

63 In addition to reducing time consumed by trains on the road, together with overtime pay to train and engine crews, large-capacity tenders effect a substantial saving by reducing the fuel consumed in starting and accelerating trains as well as the damage to locomotive machinery, draft rigging, tires and rail which frequently results from stopping and starting long freight trains. Train dispatching is simplified and the movements of superior and opposing trains are expedited, as a train which keeps moving interferes less with the movements of other trains than one which must stop frequently, thereby introducing uncertainty as to how long it will be detained.

CONCLUSION

64 The foregoing comments deal with some of the numerous details which the locomotive designer has to consider in attempting to produce a locomotive to meet practical conditions. It is to be

hoped that they will lead to a liberal discussion in order that the light of wider experience may be thrown on the subjects mentioned and in order that experience with other details may be presented for the information of all concerned with locomotive design and construction.

DISCUSSION[1]

FORREST E. CARDULLO. The locomotive, like many other mechanical devices, has not changed in fundamental construction for practically one hundred years. There was a time when no competent engineer would question the fact that the locomotive was the best possible device for the service intended. With development in other lines of engineering the question has been raised whether or not the design of locomotives should be radically changed. Many of the arguments against these changes which were valid fifteen or twenty years ago have no standing today.

The locomotive boiler was the best available for the service when the torpedo boat first came into use but with advance in marine engineering better boilers have been developed.

Radical changes in design should be tested in order to determine if there are advantages in the re-arrangement and re-design of some of the more important parts of a locomotive.

C. C. TRUMP. It seems to the writer that the time has come for radical changes in locomotive design. What objections are there to changing the present type of boiler to the water-tube type, provided such advantages as higher pressure, superheat and compactness can be obtained? What are the objections to the use of three or four cylinders in order to obtain better balance and greater tractive effort? Are the objections to these things sufficient to outweigh the advantages gained?

H. WADE HIBBARD. The writer wishes to call attention to the use of steel castings in locomotives. Objection has been made to the weight of steam-cylinder and saddle castings when made of steel, but as the designs appear to be the same as those using cast iron, apparently no advantage of the different properties of cast steel has been attained. In the same way cast-steel locomotive

[1] Many of these discussions apply also to paper No. 1797.

frames have been designed exactly as the wrought-iron frames which they supplanted. It should be possible to make a cast-steel frame of relatively greater strength and rigidity by using a box section instead of the present framing with all its cross bracing.

H. RUBENKOENIG mentioned the important part played by the engineering departments of the technical schools having courses in railroad mechanical engineering, such as that at Purdue University where much research in this direction has been conducted. It seemed to him that no theory and all practice would never produce results, nor would all theory and no practice. A combination of theory and practice resulting from coöperation between technical schools, the railways and manufacturers of the railway equipment would result in the good of all concerned.

O. C. CROMWELL said that more attention should be paid to the design of steel castings in order to obtain better results through their use. The metal should be better distributed and the edges beaded at points where the strain is concentrated. He was of the opinion that more attention should be given the proper heat treatment of the casting in order to improve its physical properties, increasing the tensile strength and elastic limit, and to elminate the internal stresses, thereby getting the maximum service out of the money expended.

He said that the frame of the box section, such as would combine the two frames in one casting, was desirable, but offered difficulties in the way of design which are interfering with the machinery located between the frames.

Broken frames can be very largely reduced, he said, if the mechanics will pay proper attention to keeping the binders tight and in place. He thought the author was correct in his introduction of cross braces but he would apply the braces in such a way as to bolt them more securely to the frame, and design them so as to have a longer fit or bearing lengthwise of the frame.

R. W. BROWN. Mr. Haig's remarks about restrictions and limitations imposed on locomotive construction are very true. The large locomotive has been greatly restricted in the cab and in the dome and the length and capacity of tanks of many of them are rather small, due to the enginehouse length, turn-table and clearance restrictions.

In Par. 20 the cracks on driving-wheel spokes between the hub

and the counterbalance are referred to. While some such difficulty has been experienced on the N. Y. C. locomotives, more difficulty has been found with cracks in the spokes between the crankpin hub and the rim. The section through these spokes has been increased but here is an opportunity for the manufacturer to coöperate in obtaining good castings. While upon the subject of driving wheels, there have been many cases of porous driving-wheel hubs on the inside of the wheel, it being necessary many times to face the hub and apply a boiler-plate liner in order to provide a clear surface for the bearing between the hub and the driving box. This defect is strictly up to the steel foundries.

While the construction of the union link connected to the crosshead pin appears to have many advantages, there is the question whether it is as satisfactory as the long combination lever. Where the union link is connected to the crosshead pin, the combination lever must necessarily be short and the distance between the pins at the top of the lever is so short that lost motion seems to accumulate more rapidly than it does with a long lever. Besides, the connection to the crosshead pin complicates the construction of the pin and the facilities for lubricating it, especially where grease lubrication is used.

A record of broken piston rods on the N.Y.C. indicates that while many breakages occur at location No. 1, there are also a great many which occur just outside the crosshead fit or in the fillet between the body of the rod and the crosshead fit. Very seldom does a piston rod break at point No. 4. The sketch shows a shoulder on the rod at the end of the crosshead fit. This design is not used on N.Y.C. locomotives. At one time piston rods were made with a shoulder at the end of the piston rod in the crosshead fit, but on account of the difficulty in obtaining a good taper fit and one that would draw up tight and at the same time register against the shoulder, the shoulders were omitted entirely.

Another contributing cause to piston-rod breakages is a small fillet where the body of the rod joins the crosshead, and to help overcome failures at this point, a 2–in. radius has been adopted.

Mr. Haig's suggestion that all sharp corners around the keyhole be rounded or chamferred is good and will undoubtedly prevent cracks starting, and further, when the piston rods are turned or ground, care should be taken to prevent sharp corners or shoulders being left on the body of the rod at the crosshead end.

The design of frame braces, especially the bolting surfaces,

is important. Cored pockets in the braces where they bear against frames or other parts should be avoided so that where the brace is bolted to another part, a full bearing area will be provided. There are many cases where the bolt fit through a large section is two or three times greater than the length of the fit in frame braces, and the bolts will remain tight in the large section and become loose in the brace. The length of bolt fit through the frame brace one and one-half times the diameter of the bolt is a good rule to follow.

It is interesting to note the comments about boiler braces, and different methods suggested to avoid cracks in the shell. Similar difficulties have been experienced on N.Y.C. locomotives and loose connection consisting of the application of a wearing plate around the belly of the boiler, upon which the boiler brace angles rest without any rivets or other fastening has been resorted to.

All that Mr. Haig has said about cab equipment, beginning with Par. 50, is true, but here is a big field for the designer. Usually, the application of many of the cab fittings is not covered by drawing so that most of them must be applied when the locomotive is nearly completed and it is too late to work out the most satisfactory arrangement. There is altogether too much congestion inside the cab and some plan ought to be developed to reduce the number of pipes in the cab, perhaps by combining some of the parts; for instance, air-brake parts in manifolds to fit the space in which they are to be applied. This, of course, would be rather tedious and difficult work for the draftsman, but it ought to be done instead of waiting until the lococmotive is nearly completed. The fact that two water glasses and the gage-cock column are now required adds to the congestion and attention should be given to these parts so that they will be handy and convenient to the enginemen and occupy the mimimum amount of space.

Mr. Snyder has presented a very interesting paper [1] and one which is difficult to criticise. His comments in Par. 13 concerning the engine proper, in which it is suggested that a design of cylinder might be considered to reduce the weight, permitting additional weight of other parts and thereby increasing the capacity of the locomotive, is worthy of note. The writer thinks that this might be desirable, but on present types of Pacific and Mikado locomotives, it is more important to reduce the weight at the

[1] See paper No. 1797.

rear end of the locomotive, and this will have to be done before a reduction in the weight at the front end can be considered.

JOHN E. MUHLFELD. The report of the Interstate Commerce Commission for 1918, covering 172 carriers operating 230,717 miles of railway and 63,108 steam locomotives, shows total gross operating revenues of $4,850,991,008, of which $1,350,623,955 or about 28 per cent, was expended on account of steam locomotives as follows:

Locomotive fuel	$496,421,664
Locomotive repairs	433,377,499
Locomotive enginemen	262,982,394
Enginehouse expenses	117,513,124
Locomotive water	24,842,550
Locomotive other supplies	9,323,584
Locomotive lubricants	6,163,140
Total	$1,350,623,955

all of which shows the great responsibility, directly and indirectly, of the railway officers and employees connected with locomotive design, equipment, upkeep and operation.

The author places particular emphasis on providing, first, " bridges, rail and road bed capable of carrying a static wheel load of at least 65,000 lb. per pair and of permitting the additional stresses resulting from a freight speed of at least forty-five miles per hour," and second, " features which have not been so generally discussed and exploited which keep a locomotive in service a maximum length of time, reduce engine failures to a minimum, reduce cost of maintenance and repairs, and increase revenue earning power."

With respect to the first item, a static rail pressure of 65,000 lb. per driving axle, will, due to the balancing of reciprocating parts, be increased, at a speed of 45 miles per hour, about 10,000 lb. when the reciprocating parts are made of ordinary open-hearth steel, and about 5,000 lb. when such parts are made of light high-grade alloy steel. Therefore, it is of the utmost importance that every effort be made to reduce the weights of pistons, piston rods, crossheads and main rods, and that exceedingly large freight locomotives should be designed so that the piston thrusts and stresses and the reciprocating weights will be distributed, as in the Mallet, rather than centralized, as in the Santa Fé types, in order that excessive total rail pressure shall not be produced by any

one pair of driving wheels, and further for the reason that any reduction in excess balance will permit of the utilization of a greater static weight upon driving wheels without detrimental effect on rails and other track appliances, or on bridges.

In regard to the second item, the writer will not discuss the various details on which the author comments, but will confine his review to the substance of the paper which, in his opinon, is most opportune and timely.

During 1920 the Class I Railways of the United States, or those having gross operating revenues of over $1,000,000.00 per annum, represented 231,456 operated miles of road and over this mileage was moved 914,224,232,000 freight gross ton-miles by approximately 39,000 freight locomotives which made 720,407,000 miles. These freight locomotives each averaged about 18,500 miles during the year or but little more than 50 miles per day, whereas they should be capable of averaging at least double that amount, or 100 miles per day. In fact the total passenger, freight, mixed, special, switching and work-service locomotives averaged only about 27,500 miles each during the year, or about 75 miles per day, which is but little over three miles per hour. Add to this low productive capacity the fact that, comparing 1920 with 1915, the cost for new locomotives has increased from eight to twenty cents per pound of light weight, or about 150 per cent, and that the average wage of all railway employees has been increased 130 per cent with an additional increase of 32 per cent in the average total number of employees, and there is little wonder that the return on railway property investment has gone down from the already low figure of 4.17 per cent in 1915 to 0.33 per cent in 1920.

During the five-year period, 1916 to 1920 inclusive, about 10,500 new locomotives, or an average of 2100 per year, were put into service in the United States, principally to replace existing locomotives that had become generally obsolete and inadequate, and therefore uneconomical for operation and maintenance. With the continuing high interest rates and cost for new locomotives, $350,000,000.00 of railway funded debt, exclusive of equipment obligations, to be refinanced this year, non-payment of large amounts of money due the roads from the Federal Control and Guaranty Period operations, established wasteful labor classifications, rules and working conditions, high labor and material costs for maintenance and operation, and more than sufficient motive power available to meet all transportation requirements, the only

justification for new purchases will be for the reason that the author has brought out, i.e., to increase earnings by more economical production. However, before these new purchases can be made, the railroads, with a few notable exceptions, must reduce present operating ratios, and reëstablish surplus earnings to create credit and borrowing capacity in order to provide capital funds. This can only be done by a substantial reduction in the present fuel, labor and other transportation costs, as well as in upkeep, labor, material and supplies per unit of work performed, and many suggestions are now being advanced as to how this can best be accomplished.

Those means and methods which will increase existing locomotive starting power and hauling capacity and promote fuel and other economies without any considerable capital outlay should receive first consideration and these include many possibilities such as, for example:

1 Reduction of abnormal factors of adhesion by increasing working steam pressures from 5 to 20 lb.

2 Increase in superheat by locating superheater element rear return bends within 12 inches of firebox flue sheets, when repairs and renewals are made and by increasing superheater flue gas areas

3 Application of arch tubes in fireboxes spaced from 10- to 14-in. centers in order to provide economical sectional fire-brick lengths, increase effective firebox heating surface and improve boiler circulation

4 Revision of locomotive ratings in order to utilize increased capacity provided by foregoing changes

5 Distribute locomotives over divisions and districts so as to insure maximum percentage of gross ton-miles hauled to locomotive rating ton-miles

6 Purchase of most suitable materials for renewals and repairs involving large labor expenditures, and proper inspection and testing of same prior to acceptance

7 First-class enginehouse maintenance of cylinder and valve packing, boilers, superheaters and flues, exhaust-nozzle and draft appliances, brick arches, grates and ash pans

8 Establishment of adequate and effective central and field supervision and records to follow up, by personal contact with engineers, firemen, enginehouse men and locomotives the work of handling, inspection, maintenance,

preparation and use of locomotives and fuel and supplies at terminals and on the road to reduce wastage.

The author has stated that the proper performance and development of the steam locomotive "requires the unlimited cooperation not only of the mechanical, civil-engineering and operating forces of the railroads, but also of the locomotive builders and particularly of the manufacturers of material." In this the writer heartily concurs, as until many existing, obsolete and wasteful ideas and practices which are being continued by some of the controlling designers, manufacturers and builders are changed, the necessary relief will not be forthcoming, and otherwise the introduction of the more efficient new locomotives and special equipment pertaining thereto will be delayed and use of the existing motive power will be continued.

In other words with adequate motive power available to carry on the business of the country, nothing is to be gained by the purchase of new, as a substitute for present, locomotives of like tractive power, unless the resulting economies will justify the cost of the new and the retirement of the existing.

C. J. MELLIN. The author treats the subject of the designing of present-day large locomotives and brings out some important points for consideration, and otherwise in the main runs parallel with previous papers on locomotive construction presented to this Society.

Among the points referred to he recommends the reintroduction of the Laird crosshead, which was practically abandoned twenty-five years ago because of an epidemic of broken piston rods attributed to its use, so that the Alligator type of crosshead has come into general if not exclusive use.

For slow-speed engines however, (if always run at moderate speed) the Laird crosshead is all right as it has the advantage (in addition to its lighter weight) of the location of the guide well above the ground and is less exposed to sand and dust than the lower guide of the Alligator crosshead.

The counterbalancing of the driving wheels has always been a question for controversy and although no fully satisfactory rule can be laid down, a kind of average has been agreed upon, namely the A.R.M.M. rule of 1915 mentioned by the author, that cannot be properly applied to large engines with small wheels, and only

by a subdivision of the driving forces at different crank angles to each other can a more satisfactory counterbalance be obtained.[1]

The bracing of the frame is one among many other important points of locomotive design and the introduction of the Walschaerts valve gear has made it possible for this to be very satisfactorily done, but it seems to the writer that the design submitted is somewhat lacking in tram bracing for so large an engine.

A. W. Bruce. Locomotives are the source of a railway company's revenue, and should be designed

1 For the work to be performed
2 Minimum maintenance
3 Maximum economy (both fuel and weight per lb.)
4 Least wear and tear on rails and other equipment.

Most important of these is the work to be performed. We have much information in general about the other items, but too frequently very little information as to the service expected of new engines.

The railway companies should, whenever possible, confer with the locomotive builders, giving all characteristics possible of existing road, and peculiar conditions, and after getting the engine designed to meet these conditions, the engine should be kept on this division as much as possible.

The general-purpose engine is desirable, but will not always give the best efficiency at all times, nor could this be reasonably expected.

The quality of material obtainable largely determines design and service of engine. The designer can do nothing further than is permitted by the physical characteristics of his materials. This in turn is a commercial question as shown in the automobile industry.

We could probably obtain better materials were there sufficient demand and were we willing to pay an increased cost. The designer of new power must remember that railway companies are very likely to replace any repair part with material from stock which may or may not be equally as good as the original part. Much has been done toward simplifying the material situation during the war by the adoption of the U.S.R.A. Material Specifications. It is to be hoped that a common standard for material may be adopted by the different railroad systems, too many of which have in the past

[1] See also discussion by C. S. Mellin of paper No. 1797.

adhered strictly to their own specifications which differed only slightly, but sufficiently to call for several grades of material.

Engine failures only too often occur from faulty small parts, such as pipe connections, fittings, etc. Too much attention cannot be paid to these minor details which should include particularly grate and ashpan work.

Counterbalance weights have always been a source of trouble. The balance of reciprocating weight is put in to make a steady-riding engine and anything increasing this quality would perhaps permit slight reduction in the necessary counterbalance.

It would seem then that, on long wheelbase engines arranged with constant resistance trucks, the counterbalance for reciprocating parts might be slightly reduced.

The Laird type crosshead was popular in 1900. Little was seen of this crosshead from 1910 to 1920. What are the peculiarities of this crosshead which are bringing it back in service?

Cylinder design has now become standardized and fairly satisfactory. On account of foundry difficulties, cast-steel cylinders have not been successful. Has anything been done toward overcoming this?

Cast-steel driving wheels are now almost universal on heavy engines. What experience has been had on adjustable hub liners and different thicknesses of tires? Also, has there been much trouble of late due to axle failures?

It seems to the writer that the breakage failures of these items is largely a question of maintenance. No amount of good design or material can obviate trouble if wearing parts are neglected.

Proper maintenance, shoes and wedges, boxes, rods, etc., will, the writer believes, go further than anything else toward keeping engines on the road. It seems to the writer that proper maintenance, even with present shop conditions, is the cheapest method of maintenance.

It might be well to question experience with floatnig bushings on main pins for main and side rods. The writer believes this device, if properly lubricated, gives good satisfaction.

Frame bracing is, of course, very important. Especially considering the difference in section between the main frames and the braces attached. Also consider the fact that the front deck casting is placed between the frames, the cylinders fit outside the frames; guide and crosstie between the frames. Is it any wonder that trouble is experienced in frame alignment. If properly aligned

originally, and if shoes and wedges are properly maintained, the writer believes frame breakage conditions will improve.

The boiler should, of course, be kept as free as possible from rigid connection to main frame.

Fireboxes are now getting to be very large in size, and should be made with crown and sides in one piece whenever possible, avoiding seams, etc. Ashpan and grates should be well proportioned and kept up to avoid failures.

Particular attention should, of course, be given the fuel used.

Cab arrangements on large engines are becoming complicated affairs. Cab turrets and reverse gear have been placed outside of the cab. Stokers, and flange lubricators, however, are taking their place so that no additional room is obtained. It might be well to consider cutting away the front part of cab on large engines where no front door is obtainable in order to provide accessibility to firebox staybolts.

As a matter of fact, on large engines the engineer has very little chance to see ahead of the engine without leaning out the window. Opinion might be asked as to the advisability of locating the cab at the front end of the engine, as has been done on the Southern Pacific Mallet engines, using oil fuel.

This is, of course, a logical location for the engineer's cab, but if coal fuel is used an additional man will be required on the engine.

Tenders have also been a source of trouble, but now that cast-steel frames are being used, etc., maintenance is much less.

The question might be asked as to the experience with six-wheel trucks, and, as to the size of tenders which may be expected in the future.

J. G. BLUNT. It is evident that the locomotive sought for by motive-power officials is one that enters into and remains in service continuously with the minimum of repairs, maintenance or attention, and while improvements suggested with the idea of saving fuel, increasing hauling capacity and reducing maintenance costs are many, they should be obtained by means of the simplest possible design embodying the fewest number of parts that would produce effective results. Designs so produced are usually the most economical to manufacture and maintain; at the same time most difficult to obtain. Thoughtful and experienced experts should, therefore, pass on new designs as well as improvements to old ones,

recognizing that a result may usually be obtained in countless different ways, and even expert judgment often fails in obtaining the best results with the least expenditure of labor, material and maintenance.

Experience has shown us that a thoroughly designed locomotive requires the attention of specialists responsive to manufacturing and operating needs. This applies to detail design, calculations for strength, curvature, centers of gravity, library references, government regulations, service records of locomotives in use, etc., and generally the fewer numbers of parts individually specialized on, the more perfect the product. Therefore, design developed on these lines should represent less investment by the railroad, less overhead for maintenance and more tonnage hauled per locomotive in a unit of time. This has been most apparent in the manufacture of standard machines where the workman often specializes on a single unchangeable detail, resulting in marvelous accuracy.

Railroad officials are and have been very helpful by encouraging designers in the use of improvements to overcome existing weakness, reduce maintenance costs, eliminate useless parts and in simplifying construction. Yet in summing up a proposed improvement from a final profit or loss standpoint, the question of carrying added stock for repairs by the railroad must receive consideration.

Standardization of such details as lend themselves to uniform use by railroads and locomotive builders has been a live subject for many years and while there are many limitations to its wide extension, yet it is important from the standpoint of the railroad official responsible for keeping his motive power in service the greatest percentage of time, as well as the designer and manufacturer, from the fact that his methods, materials and designs lend themselves to vast improvement if his energies are centralized on the perfection of standard details, rather than attempting to obtain equal perfection in a great variety of designs developed for the same purpose. Illustrative of this point, it would seem engine truck axles, boxes, and bearings might with propriety receive the same general standardization as have tender axles, boxes, and bearings.

Other important considerations, are the quality of materials which must be made to suit work that has to do with resisting stresses, forming bearings and conditions for the resistance to corrosion, also the treatment of the material, which makes it best for the requirements.

It would seem safe to say that the average locomotive receives

lubrication only at such points as would cause trouble if not lubricated. Yet an important feature in the operation of a locomotive not always receiving the careful attention it should is the lubrication of minor parts, some of which are provided with openings to receive lubrication, such as spring rigging, but in many instances do not receive the lubricant they need to prolong the life of the parts. Doubtless the means provided, as well as attention in service could be improved to advantage. It is possible that many working parts now providing for no lubrication, such as drawbar pin bearings, and in many cases center plates, side bearings and the like, would respond surprisingly if treated with a lubricant, both in wearing qualities and in the reduction of frictional resistance, thus increasing hauling capacity and reducing maintenance. The designer may lend assistance in overcoming these difficulties by providing in the design for cavities of sufficient capacity and so located as to provide a continuous supply of lubricant requiring only periodic attention.

High-class workmanship and inspection should accompany similar efforts in design to obtain a highly effective product, and railroad representatives and manufacturers of special parts working together with the locomotive designers, shop superintendents and inspectors, have worked wonders in the way of development and effectiveness of locomotive details. The fact that so many individual parts of the locomotive have been combined into single units, both in the larger and smaller details, shows what can be done for simplicity and safety.

While the two papers presented by Messrs. Haig and Snyder[1] have enlivened interest in so many ways, it has often occurred to the writer that a frame, to best resist the combined stresses of known as well as unknown character, should follow as closely as practical the lines of a tube, and in designing a locomotive to pass curves and to respond to vertical track inequalities should, wherever practical, in adjusting itself thereto, be restored to its normal position by pressures exerted upward in the ordinary spring rigging suspension rather than adding other centralizing devices which tend to complicate the general construction.

As to the subject of boiler design, the problem of the designer seems to be largely placing the minimum obstructions to the free and unresisted expansion of the boiler plates. This obstruction largely results from the application of staybolts, boiler braces and

[1] See also paper No. 1797.

frame-bracing attachments, and realizing that the boiler must form the real backbone for the frame, the frames should be so firmly attached to the boiler as to form an immovable attachment for the guides and other machine working parts, together with the varying temperatures to which it is subjected, show conclusively that there are boiler problems which alone have baffled the past-masters in locomotive design and still engage the best minds of today in their attempted solution.

Safety to life and property exert a pronounced influence in locomotive design. This requires the closest analysis to proper resistance in passing curves at speed, reduction of the dynamic augment in driving-wheel counterbalance to a minimum, accurate registry of boiler conditions as to water and pressure. The general use of the water column has apparently come to stay, but its construction of steel tubing may well be considered in place of cast material, as well as the placing of all live steam pipes outside the cab.

Fuel economy, one of transportation's greatest problems, has resulted in the development of many experimental locomotives, some of which have been constructed; others are receiving much consideration, such as turbine locomotives, locomotives built for much higher steam pressures than heretofore utilized, internal combustion locomotives, etc. These types, in combination with compounding, superheating, condensing, feedwater heating and the further utilization of exhaust heat and pressures, together with more perfect combustion, suggest problems of a world-wide character, likewise inviting the highest order of engineering talent in their practical solution.

W. E. Symons. The design or type of crosshead shown in Fig. 1, of Mr. Haig's paper has given excellent service, and the suggestions made by the author with respect to detecting defects in the casting are very good. Steel castings for crossheads or other parts of locomotives requiring high tensile strength can be tested for inherent defects, cracks resulting from shrinkage strain, etc. by first immersing them in benzine or kerosene, then after thoroughly wiping them, removing all benzine or kerosene from the outer surface, applying a coat of white paint or whitewash, following which defects will shortly be visible, if any exist.

The type of crosshead known as the Alligator has in some instances been used in place of the Laird, with very good results,

and particularly so with respect to the manner of fastening to the piston rod, as shown in Fig. 4, with particular reference to the key and key-way, as shown in detail in Fig. 5, and while the writer favors the use of the Alligator crosshead, with double nuts on the outer end of the piston rod in the place of a key as used in the Laird process, he would suggest to those who prefer the latter that they look sharply to their standards of shop practice with respect to fitting up and the final assembly of the crosshead and piston rod.

An improper fit of any of these parts is what is known as a hidden defect, for when the key is once driven home no one knows whether it is a proper fit or not except the mechanic who did the job, and frequently he does not know that the slot or key-way is not in alignment, and that in driving the key home, the end of the piston rod is frequently mutilated and sometimes almost pried off or torn asunder, and this information is never known, or seldom ever known, until a piston takes out the front cylinder head, possibly tearing a piece out of the cylinder and not infrequently doing other damage.

Some years ago the writer was confronted with an unusual number of engine failures. On investigation it developed that many of these failures were caused by broken piston rods through the key-way in the crosshead fit. Positive instructions were issued that no machinist, no matter how competent or reliable, should be allowed finally to drive home a crosshead key until after either the gang foreman or general shop foreman had given it a preliminary test drive and certified over his own signature as to the accuracy of the work, and following the enforcement of this rule there was no more trouble with piston rods breaking in key-ways.

In the matter of counterbalancing, the author's points are well taken, particularly with respect to high-grade material; in fact it is quite essential that special grades of steel be used on all high-speed modern locomotives, particularly in the revolving and reciprocation parts.

The question of counterbalancing has been largely solved in the use of the four-cylinder balance compound engine, with which the author of this paper is quite familiar. The writer recognizes, however, that it is a more complicated machine than the two-cylinder engine, that it is not well suited for our heavy freight service in American practice, and it is therefore quite proper that railroad companies who do not use it on their passenger and high-speed freight trains should devote their energies toward a higher

degree of refinement in counterbalancing our present standard engine.

In Par. 55 the author speaks of lubrication, which is a very live subject. Would it not be well to consider the results obtained from the use of power or force-feed pumps for the lubrication of locomotive cylinders and valves, particularly so in view of the fact that on some lines of railway cylinder packing rings are replaced after 20,000 to 30,000 miles of service, and not infrequently at a lower mileage, while superheated engines with a force-feed lubricator have been known to make 90,000 to 100,000 miles with one set of cylinder packing without the rings or cylinder walls having shown any appreciable wear.

Referring to Mr. Snyder's paper,[1] many of the points mentioned by the author, which are very excellent ones, are so closely related to the same or similar points in Mr. Haig's paper, that an analysis of, or answer to, one will frequently serve as an answer to the other.

No one who has investigated will dispute the value of many of these modern devices which make our locomotive of the present day so efficient, and most all will agree that in the construction of new power they should be considered as part of the completed unit, and that many engines now in service, which are comparatively new, could be much improved in efficiency by the application of one or more of these tried-out devices of known merit.

One particular device, known as the booster, provides a means for giving either a passenger or freight locomotive additional capacity when either starting from a station, pulling out of a yard on a reverse-curve, or getting over the peak of a hill at the top of a grade, by utilizing for adhesive purposes the weight now supported on the trailing wheel, which would ordinarily be equal to furnishing a switcher or pusher engine of about 15,000 lb. tractive power.

On a heavy Pacific-type engine handling a local passenger train with numerous stops in a busy district, this booster alone should result in giving the engine increased capacity following each start from the station, which would result in a saving of from one to two minutes of time. Therefore, on a given piece of track where a train was coming in fifteen to twenty minutes late every day, assuming fifteen to twenty stops on the division, the same engine with this device applied could bring the train in on time, with time to spare.

[1] See paper No. 1797.

Its value in handling heavy through passenger trains, while not brought in requisition quite so often as in local service, would nevertheless result in materially increasing the efficiency of the locomotive.

When one considers that a local passenger train of say 500 tons, running sixty miles per hour, has 5,280,000,000 ft-lb. of energy to be absorbed by the brakes in making a station stop in less than thirty seconds, and that the same number of foot-pounds of energy must be developed by the locomotive in accelerating this train to a speed of sixty miles an hour, the great value of this booster device in getting the train away from the station quickly can be readily understood. The heavier through trains of 1000 tons running at the same speed would have 10,560,000,000 ft-lb. of energy to be absorbed by the brakes in making the stop and developed by the engine in again accelerating this train to speed.

The question of variable exhaust nozzles should be given favorable consideration. Inexpensive devices can be applied, that by a simple movement of the engineer in the cab, can change the size of nozzle so as greatly to increase the power of the engine.

H. B. Oatley. The papers by Messrs. Haig and Snyder,[1] dealing respectively with the design of large locomotives, and with the question of the necessity for improvement in the design and operation of present-day locomotives, are extremely interesting, as setting forth the views of the progressive railroad mechanical official, and the forward-looking locomotive builder. It is pleasing to find that in both papers tribute is paid to the continued efforts of the engineering organizations of the supply manufacturers in contributing, through improved and carefully studied out appliances, to the greater earning capacity and economy of our American locomotives.

In both papers emphasis has been laid upon the necessity for greater strength and decreased weight of various locomotive parts, as well as upon the need for more analytical study of proportions, in order that more power per unit of locomotive weight may be obtained within the almost immoveable clearance limitations now existing.

Of the many well-presented points referred to in the papers, the questions of boiler proportions, superheater design and the effect

[1] See paper No. 1797.

that changes in these two factors produce on the drafting conditions of the locomotive may warrant additional discussion.

In the large modern locomotive a tremendous quantity of gases must be entered through the ashpan, grate and firebox, carried through the boiler and finally expelled from the smokebox through the stack. The power for accomplishing this is obtained from the exhaust steam, and the question of the most efficient use of this steam is a compelling necessity. It is safe to say that less attention has been paid to the more economic production of front-end draft, than to any other detail connected with locomotive design. It is true that the removal of the lower part of the steam pipes to a position outside of the smokebox has been advantageous, but it was largely prompted by considerations other than a reduction in resistance to the flow of gases under the table plate. It is also true that somewhat greater uniformity in the proportioning of exhaust nozzles and stack diameters has occurred. Little change or betterment in the entraining of front-end gases by the exhaust jet has actually been accomplished. A number of different cross-sectional forms for the exhaust nozzle have been tried. Some of these have indicated slight improvement, while others have been abandoned, as better results have not been shown. One or more attempts have been made to eliminate the pulsating features of the exhaust, but little benefit has apparently been derived. Intensive work on this question is eminently desirable, and in obtaining these results the following factors need to be included:

a The cylinder back pressure shall not be increased but, if possible, decreased

b An appreciable increase in gas entraining areas shall be provided

c The area of exhaust pipe opposed to the flow of the gases shall be measurably reduced

d The permanency of alignment, and continued efficiency of the gas ejector shall be provided.

In a reduction of the resistance to the flow of gases through the boiler, due consideration being given to the requirements for evaporative capacity, the question of gas area between the firebox and the smokebox is of greatest importance. Reference has been made in the paper to some limitation of tube length to tube diameter. Consideration of this point is needed, and in arriving at the best results there should be a disregard for style, and for the

intangible limitations that have existed as to tube diameters. There seems to be no reason why 2¼-in. boiler tubes should be the upper limit in practical use. It is not so many years ago that ninety-nine railroad men out of a hundred considered a 2-in. tube as the largest that should be used. That condition existed at a time when few, if any, tubes were more than 15 ft. long. As the Pacific type, and a little later the Mikado type, engine came into use, tubes of 18 ft. and 20 ft. became more common, and progressive designers went to a 2¼-in. outside diameter tube. With the further increase in locomotive size, involving tube lengths of 25 ft., and even 26 ft., the 2¼-in. tube has, with one or two exceptions, been continued. It will be generally admitted that the ratio of tube length to internal diameter, in such cases, results in undue increase in the resistance to the flow of gases, and nullifies, to some extent, the evaporative capacity of the boiler unless the front-end draft is unduly increased. In order to obtain this high front-end vacuum, the locomotive power is decreased because the reduction in the exhaust-nozzle area increases the back pressure in the cylinders. A marked gain in the all-around efficiency of the locomotive may be obtained by a study of conditions.

The writer, for a number of years, has advocated, as generally desirable, relation between tube diameters and length over tube sheets as follows:

Outside diameter boiler tube, in.	Outside diameter, boiler flue, in.	Maximum length over tube sheet, ft.	
		Soft coal	Hard coal, oil and pulverized fuel
2	5½	15	17
2¼	5½	20	22
2½	5½	25	28
2¼	6	30	33

In giving consideration to this condition Mr. Snyder has brought out the fact that a fixed rule will not be advisable, as many times conditions are imposed, which justify departure from, or the modification of, a predetermined relation between tube length and diameter. It will be argued that to increase the tube diameter in a given size boiler will reduce the calculated heating surface, and increase the temperature of the gases discharged into the smokebox.

It does not follow, however, that a decrease in the calculated tube heating surface will appreciably reduce the evaporation of the boiler, first, because the evaporation per square foot of the larger tube will be increased, and second the amount of water evaporated in the boiler barrel is changed but slightly in comparison with the total, inasmuch as the evaporation in the firebox and combustion chamber is not decreased, and may be increased if the decreased resistance to the flow of gases is reflected in better combustion, and in an increased firebox temperature. As regards the increased temperature of the gases in the front end, which would affect the boiler efficiency, there is within the province of superheater designers the opportunity to increase the degree of superheat, and thereby reduce the temperature of the gases leaving the large flues. There is also the advantage in engine efficiency resulting from reduced back pressure, which is obtainable through a reduction in the resistance to the flow of the gases through the larger size tubes and flues.

In conclusion, strong endorsement should be given to the points Mr. Snyder has brought out with reference to the steam space in boilers, as well as to the effect that increased steam space has on dryer steam for the superheater. So far as the superheater designer is concerned, there is no difficulty in taking care of moisture in the steam, but it is poor economy, both in design and operation, to make the superheater perform the work that the boiler has to do. It should be a superheater and not a boiler annex. The use of steam separators, auxiliary domes, outside dry pipes and other devices, have been proposed and studied by a great many who have been charged with the problem of boiler and superheater design. The growth in boiler size has been so gradual and steady that we have scarcely noticed a shrinkage in size of dome on the larger modern locomotives, until our larger boilers suggest the domeless British engines. The absence of a dome does not, of course, mean that approximately dry steam cannot be delivered to the superheater, but it does mean that the arrangement of throttle, separator, if any, dry pipe, etc., must be such as to take care of the conditions. With the locomotive boiler growing as it has done through the past fifteen years, the crown sheet has gone up, but the height clearance has not. It is by no means uncommon to find boilers today where the water level, with two gages of water, is only 17-in. below the edge of the throttle, over which the same is taken into the dry pipe. These boilers have a water-level area sometimes 40 ft. in length by

5 ft. in width. The grade on which the engine is working, with rolling, and consequent surging of the water in the boiler, means that solid water actually splashes up into the dome, and "slugs" pass over into the dry pipe and hence into the superheater, carrying with them a very high percentage of scale-forming matter, which, under certain conditions, produces deterioration of superheater units. While the superheater is made and can be made to carry these "peak-loads," the resulting performance is less favorable than the railroad company is entitled to, and the locomotive, due to such conditions, necessarily performs less effectively than should otherwise be the case.

The foregoing should not be considered in the light of criticism so much as pointing out some of the ways in which our steam locomotives may be still further improved.

The question of improvement, either in existing locomotives or in the purchase of new locomotives is naturally affected by the amount of money which can be expended on motive power. Many things would be done were more capital available. This phase of the question was referred to by Mr. Daniel Willard in the course of testimony he presented before the Interstate Commerce Commission Investigating Committee in Washington. He drew a comparison between expenditure which would reduce the cost of operation, but which would not increase the amount of transportation, on the one hand, and expenditures which would increase transportation and give better service to the country, on the other hand. It seems quite clear, however, that his decisions and those of other railroad executives would be unanimous in approving appropriations which would result in decreasing operating expenditures, and at the same time increase the production of transportation.

Railroads manufacture transportation. This was well stated by Mr. Haig, and was emphasized very well by Mr. Harrington Emerson in a recently published article. He likened railroads to an industrial manufacturing plant which takes raw material in the form of locomotives and cars and turns out the finished product in the form of transportation. No group of men are more earnest in their efforts to turn out the maximum of product at a minimum of cost than are the motive power officials on our railroads. Conditions over which they have no control are in many cases of a handicapping nature.

In the final decision as to whether changes in locomotive design and equipment are advisable or not a thorough study must be

made of all the factors in order that the results of such decisions may make for greater safety, greater economy and a return on the investment.

MAX TOLTZ. The water-tube boiler for locomotives has been used since 1903 by the French, especially on the Algerian Railway. The necessity for rigid foundations for a water-tube boiler was argued when such boilers were proposed for marine use to supplant the Scotch marine boiler but water-tube boilers have successfully been applied in ships.

The writer believes in adhering to simple locomotives rather than developing the compound. Increased efficiency will of course be obtained by the use of the superheater and of high steam pressures.

THE AUTHOR. There are objections to the use of compound and three-cylinder locomotives in the additional parts necessary and their location in inaccesible places which increases the cost of maintenance.

The question of the comparative advantages of simple and compound locomotives is largely one of weight on drivers and clearance restrictions. When the limit of weight on a set of drivers has been reached, articulated locomotives become necessary. The advantages of articulated locomotives vary with the conditions of use; some roads find it more economical to operate with locomotives having a single set of drivers and others with articulated locomotives. The common type of articulated locomotive is the compound locomotive with high-pressure cylinders on the rear section and low-pressure cylinder on the forward section, although one road has developed a locomotive with two sets of simple cylinders.

The reason for the continued use of the fire-tube boiler in locomotive service is that the boiler meets the conditions of this service in the most practical manner. The absence of a solid foundation makes the application of the water-tube boiler difficult while the necessity of frequent washing of the boiler would require a great quantity of wash-out plugs involving considerable expense in removal and replacement as well as difficulties in washing.

The various devices designed to increase economy of the locomotives must be given practical trial in order to be sure that the economy obtained is not offset by the increased maintenance expense.

One-piece cast-steel locomotive frames of box section have been given considerable attention but most investigators have been discouraged by the problem. One foundry, the writer understands, has succeeded in casting such a frame and has exhibited it at its plant. The writer does not understand that the frame has as yet been applied to a locomotive although one road has expressed a willingness to make the trial. The chief objection to the one-piece frame is the lack of facilities in the repair shop for its care. The introduction of such frames would necessitate the introduction of machinery to take care of repairs to such frames.

While there is undoubtedly merit in heat-treating processes, there is also the question of the possibility of cracks and other defects developing as a consequence of heat treatment.

No. 1797

THE NECESSITY FOR IMPROVEMENT IN THE DESIGN AND OPERATION OF PRESENT-DAY LOCOMOTIVES

By H. W. Snyder, Lima, Ohio
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In view of the demands of constantly growing passenger and freight traffic, the most urgent problems confronting locomotive designers and operating officials are, in the opinion of the author, those of increasing the capacity and efficiency of the present-day locomotives. As the limit of size of cylinders and of boilers due to road clearances has practically been reached, it is evident that any increase in the hauling capacity of a locomotive without increasing its size must come about through the employment of special devices. The superheater, brick arch, and mechanical stoker have all shown their desirability in this respect. Two other devices yet to be adopted in this country are the feedwater heater and the variable exhaust, both of which have given satisfaction in Europe.

After a brief discussion of engine problems, the paper proceeds to the consideration of the design of main and side rods and crankpins to withstand the tremendous piston thrust to which they are subjected in large engines; the difficult problem of counterbalancing; frame design and cross-bracing; driving-box brasses; means for temporarily increasing tractive power on critical grades and for starting heavy loads; ash-pan design; and lubrication.

In conclusion, it is stated that the present need is to apply the many labor-saving and capacity-increasing devices which have already been worked out and are giving satisfactory service, and at the same time look forward to the possibilities of applying other devices which have already proved that they are well worth consideration and are of sufficient importance to warrant their adoption.

NO ONE, it is believed, will dispute the fact that present-day operation of high-power locomotives is one of the most vital questions with which our railroads are concerned. The demands of constantly increasing passenger and freight traffic have brought about a constant increase in size and power of our locomotives.

2 It has not been so many years since an engine of 25,000 or 35,000 lb. tractive power was sufficient to take care of all require-

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ments. Twenty years ago cylinders 22 or 23 in. in diameter were considered as about the limit in size. Constantly increasing demands on motive power since that time have brought us to the huge Mallet engines with tractive powers of from approximately 150,000 to 175,000 lb. Our simple engines have increased from 20-in. or 21-in. diameter cylinders up to 31-in. cylinders, with a tractive power ranging from around 35,000 lb. to 83,000 lb. In view of the rapid strides that have taken place in increasing the size and power of locomotives within the last few years, it seems rather out of place to predict that the maximum has been reached. It is also true that the use of improved devices has made possible the satisfactory operation of the large locomotives of today. Everything seems to indicate that we have not reached the maximum capacity of the locomotive even within the present limits of clearance and rail load, and we may expect to see these same engines made far more powerful and economical by the application of devices which are now available or which are already being given serious attention.

3 In view of the foregoing, the most vital matter which confronts locomotive designers and operating officials is that of increasing the capacity as well as the efficiency of the locomotives which we have today. In many ways these problems have already been attacked and great improvements are continually being made.

4 In the following paragraphs an attempt will be made to draw attention to some of the problems which our present-day locomotives bring forth and upon the proper solution of which depends their success.

COMBUSTION AND STEAM GENERATION

5 In order that large engines may operate properly, it is of course necessary that a sufficient supply of steam be furnished to cylinders so that they can be made to produce their maximum horsepower. It is not enough to provide a given number of square feet of heating surface in the firebox and the tubes so that we may be reasonably certain that sufficient water will be evaporated to supply the cylinders. It is, however, necessary that we take into account proper construction of the boiler, necessary firebox volume to produce the best possible combustion of fuel, and the design of grates so that fuel will be economically burned to such an extent only as required by the maximum evaporation of the boiler.

6 In producing heavy motive power it has been necessary on account of prohibitive axle loads to apply a sufficient number of axles under the engine to reduce the individual axle load to within reasonable limitations. This has lengthened out the engine to such an extent that boiler design and maintenance has become a serious problem. In the first place it is necessary to design a boiler that will properly function with the other vital parts of a locomotive. At the same time the length has become such that the use of combustion chambers is a necessity to avoid a prohibitive length of tube. Large engines have been constructed with a tube length of 25 ft. and it seems that no definite rule has been established as to what the limit of length of tube of a given size should be. Experiments have been made on this subject and it has been said that the maximum length in inches of a tube of a given size should be approximately 100 times its diameter in inches. It would seem that this is as nearly correct as any general rule which has been devised, and one which can be readily followed. The author does not feel that any definite rule should be made in regard to length of tubes, for this might bring about a condition whereby other vital features of the engine would be involved in order to abide strictly to the length as noted above. Tubes 2 or $2\frac{1}{4}$ in. in diameter in excess of 20 ft. in length are questionable, and this feature should be looked into carefully before a decision is reached.

7 The advent of long combustion chambers has brought along with it the necessity for increased attention to boilers. The application of a long combustion chamber requires a large number of additional staybolts, and it would naturally be expected that a boiler of this kind would require more staybolt attention. For this reason, if for no other, there is no doubt that a proper installation of flexible stays in the firebox and combustion chamber will prevent a great deal of the staybolt trouble which has been experienced in the past. Complete as well as partial installation of flexible staybolts has met with varying degrees of success on many railroads, but the consensus of opinion seems to be that their application goes a long way toward overcoming staybolt trouble. The extreme length of firebox sheets due to the application of a combustion chamber naturally increases the relative amount of expansion and contraction of the boiler. It has been proved by experiment that if flexible stays are properly applied to the boiler when built, while they may make a slight movement during the firing up of the locomotive, after the boiler has become completely heated and steam pressure

raised these stays assume their original position. The fact that a movement of the staybolt can be shown while the boiler is being fired up shows that something of this kind is necessary, for if rigid stays were applied this movement would have to be taken care of in some other manner, which would be detrimental to the boiler. Although long combustion chambers require more attention in maintenance, this will be offset by the increased firebox volume and the resulting better combustion.

8 On account of height limitations, the height of the dome as well as the steam space in the boiler has been reduced to such an extent that difficulties are being encountered with the proper life and maintenance of superheater equipment, because too much water is drawn over through the throttle into the superheater. This is a question requiring experiment to determine as nearly as possible the minimum steam space which should be provided for boilers working on various grades. Cole in his bulletin on locomotive ratios has given figures for the minimum height of crown for different grades and lengths of firebox. This, it is believed, is the first effort made to tabulate this information and get it in shape so that it could be followed. It is thought that in most cases the minimum height that he gives will work out satisfactorily. There is, however, one point which is not taken into consideration, namely, the height of the dome. This has a great deal to do with obtaining dry steam, and in the opinion of the author consideration should also be given to the height of the throttle above the water line as well as to the steam space in the boiler. Considerable development on this subject is now well under way and we can confidently expect results of value in the near future.

9 As noted above, it seems that we have about reached the limit of size of cylinders and size of boiler due to road clearances. To undertake to provide additional road clearance on practically all of the main lines today would mean a total expenditure of money entirely out of proportion to the benefits that would accrue.

10 On account of the apparent limitations of piston thrust and road clearances, the greatest problem we have with our large locomotives today is to increase their capacity without exceeding greatly our present sizes. Anything to increase the hauling capacity of the locomotive without increasing the height and width limitations under which the locomotive must work might be called an essential capacity-increasing device. A few of these with which we are most

familiar and which have proved beyond doubt their desirability are the superheater, the brick arch and the mechanical stoker. There are possibilities of still further increasing the efficiency of the superheater without increasing the size of the boiler in which it must operate. There are also possibilities and constant improvements in the design of brick arches which lead to higher evaporation and better combustion of fuel. It has been stated that when a locomotive requires as much as 6000 lb. of coal per hour it has gone beyond the limits of the ordinary fireman. Automatic stokers have been in use so long that their dependability for heavy power is no longer in question. Many men are studying this particular feature of locomotive design and operation, and we may confidently expect in the future a gradual increase in the efficiency of these mechanisms. As they stand today they are an unqualified success, and time and study will bring about the necessary refinements so that better combustion will result and less coal per horsepower be used.

11 We have not as yet gone very extensively into the use of feedwater heaters. It is not the author's purpose in this paper to furnish facts and figures as to the amount of economy that can be obtained by the use of these various devices. Many who are engaged in a specific study of each of them will be only too glad to furnish information covering what savings can be expected by the use of these later up-to-date improvements. It has been proved without a doubt in foreign countries that the feedwater heater is an essential capacity-increasing device as well as an economical addition to the locomotive. The author has information from men in the railroad division in the late war showing that practically all of the German locomotives taken over by the Allies after the armistice were equipped with feedwater heaters. Their use in Germany has come to such a point that they apparently do not consider producing large locomotives without this device. In this respect, then, it would seem that we are somewhat behind the Europeans, and there is no doubt that in the near future when the economies that can be effected by the use of feedwater heater are realized it will become almost as general as the superheater today.

12 Another small item which has received only passing attention in this country is the variable exhaust. As is well known, a variable exhaust that can be properly operated and which will not require much maintenance attention will have a great tendency to relieve high back pressure at high speeds, and its operation will also provide

the necessary draft at slow speeds. While this may be a less important matter than many other of the points already brought out, at the same time it is one of the small things that deserves consideration and study, and something which it is felt will be worked out satisfactorily for the future. It might be said in this connection that European locomotives in a great many cases are equipped with such a device which seems to give satisfaction.

THE ENGINE PROPER

13 There have been no radical changes in the general design of cylinders. The use of outside steam pipes has resulted in advantages both from a casting and maintenance standpoint. It would seem well worth while to consider a design of cylinder by means of which the weight could be reduced to a great extent, permitting of additional weight of other parts and thereby increasing the capacity of the locomotive.

14 The design of valve gears has received a great amount of attention and many accepted types are now available. In all of these every effort has been to better the steam distribution. In maintenance we are far ahead of engines used twenty years ago. There is yet, however, much to be desired in steam distribution, and this subject will bear as careful study in the future as it has in the past.

POWER TRANSMISSION

15 When we consider that as much as 150,000 lb. piston thrust is being transferred through a single main rod and from this into the driving wheels of a locomotive, it is not difficult to understand why troubles are experienced with main crankpins and particularly side-rod bearings at the main pin. In order to provide the proper strength to take care of this tremendous piston thrust it has been necessary to design extremely heavy main and side rods. The piston thrust is not the only consideration in this connection. The inertia forces, particularly in drifting, at times reach figures that are even greater than the piston thrust. Practically all of this must be taken care of through the main crankpin and the necessary connections to the side rods at this point.

16 All are familiar with the large number of experiments which have been carried on to produce a steel that would give a higher elastic limit than the ordinary high-carbon open-hearth steel which

was successfully used until engines reached their present proportions. The use of such steel for side rods, main rods and piston rods has been principally confined to heat-treated and quenched forgings, which permitted the use of sections which were considerably lighter than what could be used with the ordinary open-hearth annealed forgings. Steel has also been produced which gives a high elastic limit and which can be successfully used with ordinary annealing, permitting very considerable reductions in weight compared to the ordinary open-hearth steel formerly employed. The use of such a steel does away with quenched forgings and permits of rods being heated for closing in straps and similar work without destroying the quality of the material as is the case with quenched forgings.

17 Main and side rods have been produced and have been in successful operation for the past few years in which the piston thrust is carried directly from the main rods to the side rods back of the main wheel. This does not in any way reduce the piston thrust that must necessarily come on the main rods. At the same time, however, it does reduce very considerably the piston thrust that must be transferred through the main crankpin into the side rods, thereby alleviating to a very great extent the troubles that have been experienced with large side-rod connections at the main pin. Such a design does not increase the total weight of the rods to an extent likely to cause any appreciable increased difficulties from a counterbalance standpoint.

18 The design of main and side rods as well as main crankpins will always be a vital question in the construction of locomotives. We have, of course, and no doubt always will have, failures of main and side rods — more with side rods than with main rods — and the problem is to reduce them to a minimum. It has been necessary and always will be in designing the rods for locomotives to assume certain arbitrary limits of fiber stress based principally upon past experience. It is impossible to take into account all the stresses produced in rods when a locomotive is in operation, and for this reason the allowable fiber stresses in tension, compression and bending must be small in comparison with the elastic limit ordinarily obtained in such forgings.

19 Rod design is a study in itself and presents a subject the details of which cannot be covered in a paper of this nature. There has already been a great deal written and a number of experiments conducted regarding the proper design of rods to successfully stand

up under severe usage and at the same time reduce to a minimum the ordinary difficulties presented from the standpoint of counterbalance. Hollow-bored piston rods, light designs of crosshead and piston, the use of high-tension steel for side and main rods, as well as the use of hollow-bored crankpins, are familiar to all. More careful attention should be paid to the quality and upkeep of rod bearings, and every endeavor should be made to provide bearings of such quality and design that renewals will be reduced to a minimum.

20 Before the advent of our present-day large locomotives with their tremendous piston thrust it was not a particularly difficult matter to design a suitable main crankpin. The size of the main pin was made to a great extent to accommodate the cylinder centers and other parts of the locomotive, and so long as the bearing pressure per square inch of projected area based upon maximum piston thrust was within a limit of 1600 or 1700 lb. the main pins would work satisfactorily. The ordinary design of smaller locomotives was such that the main side pin would also be sufficient. It has been found in comparatively recent years, however, that the old rule would no longer apply. In order for the main crankpin to be of sufficient size to withstand heavier piston thrust and still maintain the fiber stress within workable limits, it was necessary of course to increase the diameter proportionately. This brought up the question of rubbing speed. It is a well-known fact that if the rubbing speed is too high, bearings will heat and wear very rapidly regardless of the bearing pressure. A main pin designed properly for heavy piston thrust must therefore be so proportioned that the length will bear a certain relation to the diameter within very close limits. On account of the necessity for keeping cylinder centers as close together as possible because of road clearances, if a proper length of main pin is obtained, its proper size presents a difficult proposition. This is one of the great difficulties which the author is confident will be overcome in the future by the proper application of a design previously mentioned, wherein a large part of the piston thrust is transmitted directly from the main rod into the side rod, thus reducing the force that previously has been taken through the comparatively short and large-diameter main side pin. It is not sufficient to establish a definite set of cylinder centers and frame centers for a given diameter of cylinder. In the design of large locomotives different conditions and problems are confronted which make this point a particular study in itself and which should be gone into thoroughly before a decision in regard to the design is reached.

COUNTERBALANCE

21 It is a very difficult matter to separate the question of counterbalance from the design of connecting rods and reciprocating parts. There is a great diversity of opinion in regard to the proper amount of counterbalance which should be applied to locomotives. It is easier to state the counterbalance result desired than to tell exactly how it is to be obtained. We ordinarily think of a locomotive being counterbalanced properly when it will ride satisfactorily and without excessive vibration, and it seems that this is the rule by which counterbalance is judged. There are in operation heavy Santa Fe type locomotives which have between 35 and 40 per cent of the reciprocating weights counterbalanced, and they are said by traveling engineers to ride easy. The author believes that with our present heavy engines with long wheelbase it is not necessary to balance as much as 50 or 55 per cent of the reciprocating weight. In fact, it is quite possible that we may be able to counterbalance a smaller percentage of reciprocating weight than has heretofore been attempted, especially for long, heavy engines, provided the revolving weights at the main pin can be properly taken care of. Every effort, however, should be made to balance all of the revolving weights on the main pin. If, for example, we lack 400 lb. of balancing the revolving weight on the main pin, the effect on the track is exactly the same as if we had 400 lb. of counterweight on any of the other wheels to balance reciprocating parts.

22 There is an added difficulty to this problem, because the action of the counterweight in wheels other than the main is exactly opposite to the force produced by the weight on the main pin which is not counterbalanced. This condition results in increased track stresses, as well as increased stresses in frames and other parts of the locomotive. There is also a tendency at high speeds when a condition like this exists for the main wheel to lift from the rail, while the wheels other than the main are exerting their maximum force on the rail. This reduction of weight on the main wheel at the time when the other wheels are exerting their maximum force on the rail provides a tendency for the main wheel to slip when it is impossible to slip the remaining wheels. No one, it is believed, can give any idea of the stresses which are produced in side rods, frames and other parts of locomotive due to a condition of this kind. In fact, there have been instances where the rods were torn off and crankpins loosened for this reason alone.

23 On account of the extremely heavy weight required at the main pin in order to have the proper strength of parts, particularly for large freight locomotives, it is a difficult matter to balance very much—if any—more than the revolving weights at this point. This being the case, if as much as 50 or 55 per cent of the reciprocating weights is to be balanced, it is easily seen that all of the counterweight for reciprocating parts must be added to the counterweight in the wheels other than the main. Thus, in order to balance a high percentage of reciprocating weights on engines of this class, it is necessary to add counterweight to the wheels other than the main to such an extent that track stresses and riding of the locomotive at comparatively high speeds become a very serious question.

24 The author is of the opinion that no definite set rule can be established in this regard, but that each particular design is a study in itself, and wherever revolving weights at the main pin are encountered such that they cannot be properly counterbalanced, steps should be taken to provide the best means possible of reducing revolving weights at this point as well as providing reciprocating parts as light as possible consistent with strength. This of course has been accomplished in the past by hollow-boring the main pins and piston rods and by using a light design of piston head, which indicates that a steel having a high elastic limit with the proper elongation and reduction of area should be employed. The use of such steel has already proved that it can be depended upon. One of the principal fundamentals in counterbalance is to keep the reciprocating weight as light as possible.

THE RUNNING GEAR

25 On account of the large increase in the size of cylinders of present-day heavy locomotives over those used several years ago, the cylinder centers have been spread until they have reached practically the clearance limitations of the railroads; and the necessity for larger journals to carry properly the increased axle loads has caused the frame centers to be brought nearer together.

26 This condition increases very materially the distance from the center of the cylinder to the center of the frame, which of itself produces greater strain in the frame and at the same time increased pressures on the driving-box bearings as well as on shoes and wedges. In addition to the above, piston thrusts have increased from approximately 65,000 lb. to approximately 150,000 lb., and means must

be provided to properly take care of the increased piston thrust along with the increased overhang.

27 Fig. 1, it is believed, shows very clearly the increased forces that a frame must stand in order to properly take care of the tre-

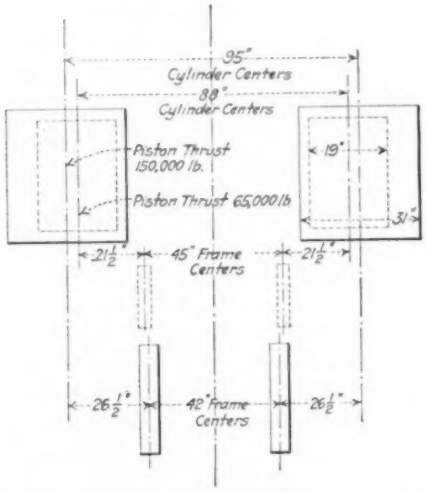


FIG. 1 DIAGRAM SHOWING INCREASE OF OVERHANG OF CYLINDERS IN PRESENT-DAY LOCOMOTIVES

mendous increases in piston thrust as well as the increased leverage caused by the very considerable lengthening of the distance between the cylinder centers and the frame centers.

28 While discussing the subject of frames it is hardly possible to ignore the vital question of frame cross-bracing. Substantial and sufficient cross-braces should be applied between the frames and rigidly bolted thereto to form a rugged structure which will not rattle to pieces. Sufficient bearing for bolts and adequate bolting flanges are a very important feature. At the same time it must be borne in mind that there is a possibility of tying up the frame so rigidly that there will be a tendency for failures ahead of the front pedestal and just back of the cylinder fit at a point where it is practically impossible to obtain sufficient reinforcement.

29 It seems as though the design of driving boxes and driving-box brasses had not successfully kept pace with the rapid increase

in piston thrust. We have in almost general use the same type of driving-box brass which has been standard on locomotives for years. The design is such that the brass extends about half-way down over the journal. Inasmuch as this brass must take up the piston thrust, it is very evident that we shall have trouble in taking care of driving-box brasses until a suitable design is produced — one in which the brass will cover much more of the front and back projected area of the journal than is now the case. On the heavy engines of today the main driving-box brasses in particular wear rapidly, due to the tremendous thrust which they are forced to stand up against. It is true that some work has been done along this line and designs produced which wlll help at least to overcome this difficulty. There is, however, a wide field for improvement and this particular subject alone warrants the study and investigation of all who are concerned in the design and operation of locomotives.

30 Many experiments have been tried, and in many cases they have gone beyond the experimental stage and have proved to be very satisfactory. Among these are driving-box brasses cast into the driving box. Many boxes are in use today where the brasses are keyed into the box, making them readily removable for replacement. However, the author feels that none of these has approached the solution to the problem, which lies in producing a design that will surround the journal with a bearing as completely as possible and still lend itself to comparatively easy maintenance and reasonable cost of application.

GUIDING AND TRAILING TRUCKS IN CONNECTION WITH LONG WHEELBASE

31 With our present heavy Mikado and Santa Fe type locomotives the length of rigid wheelbase is almost if not quite double the rigid wheelbase in ordinary service twenty years ago. It is unnecessary to comment upon the fact that it is a difficult matter to operate such engines around curves of even comparatively small degree and at the same time prevent the rapid wear of hub liners and driving-box faces, thus increasing quickly the lateral play to a prohibitive point and necessitating work in the shop to overcome it.

32 Santa Fe type engines with 22-ft. rigid wheelbase are not uncommon. Engines of this type and of this size will weigh in the neighborhood of 400,000 to 420,000 lb. When we stop to think that to move this tremendous mass of material around a 16- or 18-deg.

curve a force of many thousand pounds is required, is it any wonder that we obtain rapid flange wear and the necessity for returning tires before the proper amount of mileage has been obtained? In the majority of cases, it is believed, the force necessary to properly curve an engine of this kind has been applied at the front truck and the first driver. In most cases types of leading trucks have been used which produce a very small resistance on curves of small degree. In order to prevent rapid flange wear as well as to overcome the development of lateral play unnecessarily, designs have been produced which will give a high initial resistance of the front truck and provide a lateral motion for the front driver with adequate resistance so that some of the guiding force is transferred back to the second pair of drivers.

33 Since locomotives operate the greater part of the time on tangent track, it is necessary to have a high initial guiding resistance which will not be increased when curving. In other words, a flexible wheelbase is produced which has all the requisites of the ordinary rigid wheelbase, but at the same time will overcome many of the difficulties now encountered in an attempt to operate engines of this size and length. Many designs of trailing trucks have been produced with the idea in mind of helping to remedy the conditions which have been noted above. These of course have met in a way the conditions which it was necessary to overcome. There is much yet to be done in producing a trailing truck which will have the proper facilities for equalization of spring rigging and at the same time produce an initial guiding force which can be kept nearly constant, thus avoiding the high final lateral resistance which is found in a good many of the trailing trucks now in use. The use of friction plates to produce the necessary resistance is undesirable on account of the uncertainty of the resistance obtained. What is needed is a positive centering device whose resistance will be adequate initially, can be always depended upon, and which will not mount up to prohibitive figures under the maximum swing of the trailing truck.

34 In addition to the foregoing, some work has been done in the way of producing a design by means of which the lateral play in locomotive driving wheels can be taken up without removing the wheels from under the engine or taking the boxes off from the axles. No doubt in the near future a practical device of this kind will be produced. This is another one of the many problems which can be

worked out which will enable the railroads to keep their locomotives in service.

35 The advent some years ago of the power reverse gear overcame one of the great objections that engineers had to large locomotives. It is a fact that it is almost impossible for one man to reverse one of our large locomotives equipped with the ordinary hand reverse lever. Power reverse has come to be an essential part of engine equipment and has been found to be economical even though it may be used on a locomotive which could be comparatively easily reversed by hand.

36 Probably no one thing contributes more to the failure of side rods than the improper adjustment of shoes and wedges. If these are allowed to run loose, stresses in the side rods will amount to a very high figure and it is impossible to determine to what extent they may go. If the shoes and wedges are improperly set up, the driving wheels are very likely to be out of tram. This in itself brings undue stresses on the rods, which in time will unquestionably produce failures. The author feels that there are many cases where in attempting to overcome the failure of side rods we have deliberately increased their sections without giving due consideration to the cause of failure. Consideration of the foregoing brings us to the point of providing an adequate automatic adjustment of the wedge so that the difficulties mentioned will not be encountered. A satisfactory automatic wedge if properly applied and maintained will, no doubt, go a long way toward preventing side-rod failures.

MEANS FOR INCREASING NOMINAL TRACTIVE POWER

37 All railroads have points on certain divisions where there is a critical grade or the necessity of starting a heavy load under adverse conditions. At such places increased tractive power is required which is not needed elsewhere. We are therefore confronted with the problem of producing a device which can be set to work to increase the tractive power of a locomotive to such an extent that the critical grade or the necessity for increased tractive power to start a train under adverse conditions will be overcome, thus enabling the engine to take its full tonnage over the entire division. This device should be so made that it can be applied when necessary and thrown out when the additional tractive power is not required. Designs have already been produced wherein an additional tractive power of 8,000 or 10,000 lb. has been applied to the trailing trucks of large locomotives. There

is also a possibility of applying such a device to the tender truck, thus availing ourselves of the adhesive weight of the tender to help boost the engine over the critical points in a division. There is always present a possible potential boiler capacity which can be brought out by the use of a variable exhaust or other device sufficient to obtain rapid combustion at slow speeds.

38 What has already been done along this line may be taken as a start in the right direction. A certain amount of development work must be done in order that these necessary improvements may be made to operate satisfactorily. These problems require the cooperation of the railroads to provide the necessary means for trying out such devices which, after having been carefully considered, show that they have possibilities for future use.

THE ASH PAN

39 The question of ash pans is also one needing serious consideration. With the large increase in size of locomotives in many cases we have evidently lost sight of the importance of this necessity. There are in use a number of rules stating what the proper air opening in the ash pan should be, some saying that the ash-pan air opening should be equal to the net gas area of the tubes, and others that it should be a certain percentage of the opening through the grates. While many of these rules have in a way proved satisfactory, at the same time it would seem that to get at the question logically we should determine the amount of coal that can be burned economically per square foot of grate and then on this basis provide an ash-pan air opening that will give the required amount of air to burn satisfactorily the maximum amount of coal which is expected to be consumed. The amount of air that will flow through a given opening in the ash pan, it is believed, can be very closely approximated from the vacuum produced in the smokebox. This of course is only a suggestion, and it may be that when the question is looked into more carefully a more desirable and accurate method of determining the required ash-pan air opening may present itself.

LUBRICATION

40 Lubrication is a subject which has received much attention and a great number of combinations and experiments have been made to determine the most satisfactory method. With our present high superheat the proper introduction of oil into the cylinders and valves

of a locomotive is worth serious consideration. In many cases oil has been provided to both the cylinder barrel and the steam chest of superheater locomotive, and in other cases it has been provided to the steam chest alone. Both methods have given apparent satisfaction, but it is a difficult matter to state which is the better. It seems that if it has not already been done, experimental work should be undertaken to determine just how this best can be accomplished. For instance, one railroad may seem to get good results with one method, while another may use an entirely different method and both operate their engines satisfactorily. We do not know how much oil is used and it is difficult to obtain a direct comparison, due to the fact that no accurate data are kept in regard to the amount of oil supplied to either the valves or the cylinders.

41 It is common practice in European countries to provide a force-feed lubricator located very close to the cylinder. The ordinary method which they use in connecting up this lubricator is to provide a pipe to each end of the piston-valve steam chest. This oil supply opens directly over each end of the valve when it is in central position. In addition an oil pipe is supplied to the cylinder at its center. It is reported that by this method there is less carbonization of the oil than when it is fed into the steam pipes or into the center of the piston-valve steam chest. Whether or not this is so the author has no means of proving, but it seems logical.

42 In order to increase the tonnage which a locomotive can haul it is just as vital to decrease the resistance as to increase the power. Now it is not an impossibility to provide roller bearings for passenger cars, and there seems to be no reason why they cannot be used on freight cars. Of course this would mean very radical changes in design and a gradual displacement of present equipment, but the reduction of rolling resistance and the better facilities for lubrication which would be provided would be sufficient in time to overcome the necessary expense. All this may seem rather far-fetched, but it is at least worthy of consideration.

43 This question of lubrication may as well apply to other parts of the locomotive. The proper grooving of side-rod and main-rod brasses, or the use of babbitt inserts, are questions that should be taken up in connection with lubrication of these parts. A variety of opinions can be obtained from men operating locomotives in regard to the advisability of a number of methods which are in use at present and apparently giving satisfaction.

CONCLUSION

44 In summing up the situation, it may be said that the use of the superheater alone has increased the capacity of locomotives when compared with saturated engines of the same design to such an extent that no one would think of building a large locomotive for up-to-date railroad service without the application of superheat. This is one of the greatest strides that has been made in the construction of locomotives in the past few years. We must not content ourselves, however, with what has been done with this one device. The large locomotive of today has become a necessity and is here to stay. What we need to do now is to avail ourselves of the opportunities offered in the application of many of the labor-saving and capacity-increasing devices which have already been worked out and are giving satisfactory service, and at the same time look forward to the possibilities of applying other devices which are yet in their infancy, but which have proved beyond doubt that they are well worth our consideration and are of sufficient importance to warrant their adoption. There are many improvements yet to be made in locomotives and it behooves the operating officials of railroads as well as the leading minds in locomotive operation and design to get together and to continue to produce locomotives which in the next 20 years will be as far ahead of our present engines as our present locomotives are ahead of the locomotives that were built 20 years ago. Without the capacity-increasing devices which have been mentioned the large locomotive of today would be impossible — it could not be operated satisfactorily. Our large engines are an absolute justification of these improvements. Further developments are ready at hand and in their use lie the possibilities of still more powerful and economical transportation units built to operate within our present limitations of clearance and permissible rail loads.

DISCUSSION[1]

C. J. MELLIN. The author generalizes on certain points in the designing of locomotives very much in the same way as previous papers on the subject, but in some instances he enters partly on the means and methods by which the desired effect is to be obtained and mentions briefly the growth of the engines during the last

[1] For additional discussion see paper No. 1796.

twenty years, and hopes that a similar advance may be made in the next twenty years. He does not state, however, in what manner or along what lines this advance is to be obtained.

Up to the present, brute force has been mainly aimed at, and with that many difficulties have been met and overcome in a way, though in many cases not by any means satisfactorily, and very little attention has been paid to the economic utilization of the heat in the steam that is so successfully imparted by the superheater and abundantly produced by the stoker, of which at least one-half of the available heat for effective work is let out to the atmosphere; and this in itself offers a big field for future advancement.

One of the great difficulties with the larger ordinary engine is, as the author points out, the counterbalancing; it being impossible with the enormous weights of reciprocating and revolving parts to balance properly (especially in the larger engines) both with respect to the track and to the engine itself, as the peculiar condition exists that an engine that rides nicely is hard on the track, and one that is easy on the track rides hard. The balancing of a larger percentage of the reciprocating parts tends to make a good riding engine, but that excess of balance in addition to that of the revolving parts in most cases has to be transferred to wheels with small revolving weight and it thus may become doubly hard on the track.

The remedy for this is also a subject for further consideration by dividing up the number of driving cranks and connecting rods so that they as much as possible counteract each other's disturbances to the smooth running of the engine.

While this could most effectively be attained with a four-cylinder engine in a single coupling system with opposing cranks it meets with several objections; first because two cylinders will have to be applied betwen the frames where there is no room for the required size with our track gage and our modern engines; second, there is no room between the right and left axle boxes on the axle driven by the inside cylinders for two adequate cranks; third, as the angles of the cranks on this axle must be 90 degrees it makes a cumbersome and expensive axle, even when used on an engine with cylinders small enough for the available space; and fourth, the double set of guides between the frames makes the parts very inaccessible for proper attention.

The next best, and so far as can be seen at present the only available method is the three-cylinder locomotive with a single in-

ternal crank on its driving axle, which is simple to construct, has room for parts of ample strength and bearings of ample size and while not giving so good a balancing effect as the four-cylinder, it has the greater advantage of a better turning effect than any other means that may be suggested, as the subdivision of the turning forces is one of the first and greatest steps open for further advancement in improving the ordinary engine besides opening an opportunity for more extended utilization of the heat of the steam in its conversion into work.

This will have the effect of a substantial reduction in fuel consumption, and in engine maintenance; the engine will be lighter on the track and roadbed and there will follow an inestimable reduction in car repairs as the jerks in starting are reduced. It is less liable to slip and with the possibility of a substantial increase in boiler pressure, a triple stage of expansion may ultimately be adaptable with still further reduction in fuel consumption.

A. W. Bruce. The future size in cross-section of locomotives in this country will probably remain as at present for some time to come, the expense of changing railroad clearance being too great to warrant enlargement. Another limitation is allowable weight per axle of about 70,000 lb. which, if exceeded, gives trouble with present-grade open-hearth steel rails. May we expect better quality of steel in the rails? The question of changing cross-section of clearance is, of course, a very serious one in Europe at the present time, especially in England, and we may perhaps predict future growth accordingly. At present it seems that the logical growth is in the length of the engine as illustrated by the increase from 4–4–2 to 4–6–2 and at present 4–8–2 type locomotives, and also shown in 2–6–2 to 2–8–2 and 2–10–2 freight engines, and in Mallet engines from 0–6–6–0 to 2–10–10–2.

The great increase in size of engines necessitates very large firebox sheets which should, of course, preferably be made in one piece as free from seams as possible. The restricted steam space on these large engines should be given consideration, and we might question whether anyone has had experience with surge plates in the boiler. Also, what longitudinal position is best for dome location? European practice seems to locate the dome well forward, while in this country we locate the dome considerably nearer the firebox.

It would seem that the two-cylinder simple engines have nearly

reached the limit of their capacity; about the only alternative which we have would be perhaps the three-cylinder type involving, of course, additional complication of crank axle and modification of frame bracing. It would not seem that we are quite ready for this arrangement in this country.

The locomotive at the present time has about all the fuel-saving devices of stationary plants except condensers. It is not likely that these can be very successfully applied to locomotives. Fuel, either oil or pulverized fuel, will, of course, result in obtaining very high boiler output. The writer believes that at the present time consideration is being given to the mixture of fuel oil and pulverized fuel, the idea being to obtain greatest number of heat units in a given bulk. Whether or not this will be applicable to locomotive purpose is, of course, a question.

The question might be raised as to experience in this country in the use of variable exhaust which, the writer believes, will certainly be of advantage in some cases, also as to whether anything has been done recently in regard to fan draft, and with what success.

On heavy engines, the difference between cylinder and frame centers presents the greatest difficulty in maintaining alignment. This would be helped by use of three cylinders, but would result in complication of crank axle, etc. The use of special main-pin connections may be of help in some cases, and the writer believes the so-called "floating" bushing has been used very successfully on heavy power on the main pin for both main and side rods. If properly lubricated, this loose bushing seems to give satisfaction.

The use of special steel for light parts is, of course, very desirable, and has been recommended for a long time. Its successful use, however, is questionable until the locomotive designer can be assured that the railroad companies will carry in stock and use for replace parts the same high grade material that has been used in the original design of the engine. Should the engine be originally designed with a very high-grade steel using light weight and later be replaced with low-grade material, trouble will certainly result.

Counterbalance troubles have always been a source of annoyance. It would seem however, that on long-wheelbase engines as suggested by the author, especally for slow speed, we might be able to reduce the amount of counterbalance. We should certainly balance all revolving weights whenever possible. It might be noted that German engines have been built omitting balance for

reciprocating parts, and coupling engine and tender rigidly in order to obtain greater mass.

The driving-box question is also a serious one. As the greatest wear occurs about 45 degrees off the top center, it does not seem to the writer that a driving box entirely enclosing the axle would be of great benefit. The writer is under the impression that the three-piece box has been unsuccessfully tried in Europe. Long main boxes have been greatly used in this country and if properly maintained, offered considerable relief.

Summing up the question, however, of rods, boxes, pedestal shoes, etc., the writer believes the whole proposition resolves itself into a question of maintenance. If power is kept in first-class condition, the writer believes many of the present difficulties will be avoided. No amount of good design or good material will prevent trouble if the power is neglected and allowed to run down. Proper up-keep of rods, shoes and wedges, etc., will certainly prove the best investment. Present labor conditions are, of course, adverse to this, but no worse than conditions governing heavy repairs.

Ashpan, grates, etc., are certainly very important, and should be well looked after. We might question experience with outside hopper pans, non-freezing slides., etc.

Lubrication is also very important and is largely a matter of attention from the engine crew. We might question what the experience has been in this country with mechanical lubricators which are very commonly used abroad.

The writer cannot too strongly emphasize the necessity for designing the engine for the work intended as every division has its own peculiarities. A general purpose engine is, of course, desirable, but cannot be expected to give the same efficiency as an engine designed for a particular service. We should also consider that it takes time and study to design locomotives as it does any other power plant. We might profit by European practice, which, in nearly every case, covers long and careful study before deciding on any change in existing or obtaining new types of locomotives.

W. E. Woodard. While the locomotive of today looks like a large edition of that of the period between 1900 and 1907, it is as a matter of fact a vitally different machine. In the earlier days saturated steam was used and per unit of weight the best obtainable horsepower was 65 to 75, 70 horsepower per 10,000 lb. of

weight being a good figure for locomotives from 1900 to 1905. At present there are a number of locomotives in operation developing almost 100 horsepower per 10,000 lb. of weight. This considerable advance has been due to changes which were considered radical in 1905 and 1906.

A great many possibilities have been considered and many have been eliminated, those surviving which have best met the mechanical limitations of design and the result is that there is a possibility of further increase. But as the author intimated, the advance still continues. Such things as he speaks of, lighter reciprocating and rotating parts, lighter steel casting and the possibility of more extensive use of steel casting in locomotives will tend to a further increase in the horsepower per unit weight. If we were to design a locomotive today along the lines of the designs of 1900 to 1905 and with horsepowers comparable to those now in use, the locomotives would weigh over 500,000 lb. and could not be run on the railroad. This power is now being attained with locomotives weighing from 320,000 to 350,000 lb.

CHARLES RIPLEY. The expense of developing improved devices for locomotives has been enormous. Every railroad goes through the process of making expensive experiments with each device which shows promise of increased economy of operation. Why cannot this be done by some central organization so that the information will be accurately obtained and available for everyone? There is need for coöperation in the development of such devices, and there are four parties concerned, the manufacturer, the railroads, members of technical societies and universities. All these agencies brought together and coöperating would produce extraordinary results at tremendously decreased costs.

JOHN E. MUHLFELD. The paper reviews important data relating to the highly specialized steam locomotives and states more particularly "that the most vital matter which confronts locomotive designers and operating officials is that of increasing the capacity as well as the efficiency of the locomotives which we have today;" "that we have about reached the limit of size of cylinders and size of boiler due to road clearances;" "that the greatest problem we have with our large locomotives today is to increase their capacity without exceeding greatly our present sizes" and "that *further developments are ready at hand* and in their use

lie the possibilities of still more powerful and economical transportation units built to operate within our present limitations of clearance and permissible rail loads."

It is to be regretted, however, that the author does not set out what these *further developments* are, as the details referred to, such as superheaters, brick arches, automatic stokers, feedwater heaters, variable exhaust nozzles, the use of long combustion chambers to reduce the length of boiler tubes and flues, improved cylinder and valve-gear design, alloy-metals, hollow-bored parts, reduced reciprocating weight counterbalance, relation of cylinder to frame centers, frame cross bracing, improved driving-journal bearings, running-gear equalizing and centering devices, power reverse gears, automatic driving-box wedges, trailer and tender-truck booster, roller bearings for cars, and the matter of lubrication and of ash-pan air openings are such as have been in use, or under consideration and advisement, in connection with existing steam locomotive practice, both here and abroad, during the past twenty or more years. Furthermore these devices and appliances are more a matter of adjuncts or accessories than of an inherent part or feature of the locomotive design as a whole and which latter, in the writer's opinion, has not been given the scientific development by the designers, builders and users in the past ten years that the existing operating conditions and costs have warranted.

We have in the United States today, in round figures, practically 65,000 steam locomotives, which, other than the 2000 Mallet type now in use, are primarily an improved enlargement and rearrangement of the principal elements of the multi-tubular boiler, forced draft by exhaust steam, and direct connection between the piston and the driving-wheel crankpin of Stephenson's pioneer "Rocket" which was built in 1829, and these 65,000 locomotives represent over two and one-quarter billion pounds of potential tractive power.

To improve the performance and efficiency of these locomotives through changes in fundamentals, one of the simplest, cheapest and most expedient means, which the author has overlooked, is to increase the boiler pressure and reduce the factor of adhesion. Thousands of steam locomotives are being operated today with a factor of safety of boiler and a factor of adhesion somewhat in excess of the generally allowable limit of good practice of 4, whereas an increase in the existing working steam pressures of from 5 to as much as 15 or 20 lb. will produce a very substantial increase in the

available starting and hauling capacity, as well as considerable fuel economy, and as in good locomotive designing, factors of safety of from 5 to 8 are used for cylinders, frames, axles, crankpins, main and side rods, valve gear and other power-transmission parts subject to stress and fatigue, such parts would be able safely to withstand the effect of the proposed change.

While superheaters, brick arches, feedwater heaters, variable exhaust nozzles and improved cylinder and valve-gear designs will increase sustained horsepower and promote fuel economy, *they do not materially affect* the true capacity value of a locomotive, which is its starting power. For example, a Consolidation or Mikado type locomotive with 260,000 lb. weight on driving wheels and a factor of adhesion of 4 will have a train-starting power of approximately 65,000 lb. while the same locomotive with a factor of adhesion of 4.3 will have approximately 60,500 lb., or about 7 per cent less.

Certainly, and properly so, with existing driving-wheel load limitations, there is no reason for the factor of adhesion in freight locomotives to exceed 4.0, or 4.1 as a maximum, when the limiting friction for dry sanded rail will allow a factor of 3.0 and for dry rail a factor of 4.0, as by proper manipulation of the reverse lever and throttle valve when negotiating hard pulls it is entirely practicable to provide for a combination of speed and cut-off which will produce the most uniform turning moment and the maximum traction that the adhesion will sustain, all of which will obviate any necessity for factors of adhesion of over 4.1 and the hauling around of a lot of useless and expensive dead weight that can and should be made productive. As practical examples, in 1914 the Western Maryland Railway put into service 20 Consolidation type locomotives having 51-in. diameter driving wheels, 62,500 lb. tractive power, and a factor of adhesion of 3.55, and just recently they have put into service some Consolidations with 61-in. diameter driving wheels, 68,200 lb. tractive power and a factor of adhesion of 3.93. Likewise the Delaware and Hudson has in use Consolidations with 63-in. diameter driving wheels, 66,200 lb. tractive power and a factor of adhesion of 4.03, and the Santa Fé have Mikados with 56-in. diameter driving wheels, 59,600 lb. tractive power and a factor of adhesion of 3.7 which are being successfully operated with those ratios.

Another fundamental of great importance in the existing steam locomotive is the amount of superheat that is being obtained, and this vital factor is frequently neglected due to improper gas area

ratios, too few elements or the location of the superheater element rear return bends too far from the firebox flue sheet, as the result of which many locomotives are being run with from 50 to 100 deg. fahr. less superheat than would otherwise be obtainable. The increasing of the percentage of gas area through the generally used superheater flues from 40 to about 50 per cent of the total gas area, the location of superheater element cast-steel rear return bends about 12 in., instead of the usual practice of from 24 to 40 in., from the firebox flue sheet, so that they will act as "hot spots," and an increase in the total number of superheater elements will do much towards raising the superheated steam temperature and pressure at the steam chests with resulting freer steaming, easier firing, smarter working, more constant superheat temperature and generally increased power and economy.

While the foregoing concerns principally the increasing of the capacity, efficiency and economy of existing locomotives by means and methods readily available and involving the least capital expenditure, when the question of new motive power of maximum starting capacity, sustained horsepower, efficiency and economy arises, the problem becomes quite easy as in new designs of boilers, superheaters, and cylinders the use of 350-lb. steam pressure, which is now being adopted both here and abroad for stationary boilers, superheated 300 deg. fahr., is entirely permissible for locomotives. Such a combination of pressure and superheat will, as compared with 200-lb. steam pressure, superheated 300 deg. fahr., produce for the addition of only 18 B.t.u. or 1.3 per cent of the total heat in the steam and a rise of only 48.5 deg. fahr. or 7.05 per cent in the temperature, an increase of 150 lb. or 75 per cent in steam pressure. Such an increase in power, together with a more efficient boiler, improved combustion, better steam distribution and the utilization of waste heat will put the steam locomotive in a class by itself as the most effective and economical self-contained mobile power plant for the movement of fast and heavy rail tonnage.

THE AUTHOR. Mr. Muhlfeld in his discussion seems to think that further developments to which the writer refers should have been mentioned. The author is not in positon to give out information of this character, his idea being to create an interest in the foremost devices which would tend to their more extensive use. Mr. Muhlfeld further states that the various items mentioned are adjuncts or accessories rather than an inherent part or feature of

the locomotive design as a whole. This may be true, but what good would a large locomotive be without a number of these so-called accessories which have already proven their worth?

The question of increased boiler pressure was not overlooked by the author. It has not been proven as yet that boiler pressures exceeding 200 lb. per sq. in. are economical. The losses due to exceedingly high boiler pressures will go a great way toward overcoming the economies which higher steam pressures will bring about.

With pressure of 350 lb., boiler maintenance would no doubt be a very vital factor. A presure such as this for a locomotive is a much more difficult proposition than is encountered in stationary practice, where a number of expansions can be provided for and a vacuum used for exhaust. At the same time the greater the temperature of the steam in the cylinders of a locomotive the greater the loss of heat through the cylinder walls. This is one of the reasons why the economies shown by calculation for higher pressures and temperatures and extremely short cut-off will not prove out in practice until a means is devised to insulate the cylinder walls so that the excess losses are prevented.

In regard to the factor of adhesion the writer believes it will be found upon investigation that nearly all of our larger locomotives as designed today have a factor of adhesion very close to four. When this is reduced to 3.53 as noted for certain locomotives mentioned in Mr. Muhlfeld's discussion it will be found that satisfactory operation in this country will not be obtained.

Were we to attempt to design a locomotive throughout for a factor of safety of 5 to 8 some parts woud be utterly inadequate. Due to the impossibility of determining all of the stresses which come on locomotive parts we must in many cases allow a factor of safety much higher than the above based upon the known stresses in these parts. For example, it is a common practice to base the sizes of frames upon the maximum piston thrust. With this as a basis the factor of safety runs up to as high as 15 or 20. If it were possible to determine all the stresses in frames, their size would not be materially reduced although the apparent factor of safety might be lowered due to the fact that consideration had been given to stresses due to factors other than the piston thrust.

In the last paragraph Mr. Muhlfeld sums up the situaton as it has been stated before that with more efficient boilers, improved combustion, better steam distribution, and the utilization of waste

heat the locomotive will become the most economical power plant for the movement of fast and heavy rail tonnage.

Many points of particular value are brought out in Mr. Oatley's discussion [1] particularly in reference to superheaters and drafting conditions. It is possible that a means of draft other than by the direct force of exhaust steam will be perfected so that a larger proportion of the heat in the exhaust steam may be utilized than has been the case in the past.

Mr. Blunt's discussion [1] carries many important and well-taken views in regard to the necessity for standardization and the advisability of producing locomotives in the simplest method possible and yet produce a design that will be most economical and stand up under the service required.

[1] Included with discussion of paper No. 1796.

No. 1798

FUNDAMENTALS OF INTERCHANGEABLE MANUFACTURE

By Chester B. Lord, Battle Creek, Mich.
Member of the Society

IN DEALING with so broad a subject as interchangeable manufacture, it is well to remember that in the course of time many false traditions and psuedo-scientific conditions are built around a process or system, either glorifying it unduly or condemning it beyond its deserts. This makes it necessary to be iconoclastic and tear down that we may build upon a firmer foundation, sometimes rearranging the former material and sometimes rejecting it. Likewise, it is well to start out with certain fundamentals plainly stated, and any discussion or theory that contravenes these fundamentals is useless for our purpose. True, we may dispute the fudamentals and possibly disprove them, but the statements hold until that shall have occurred. Personally, however, the author disbelieves in fundamentals, and prefers to employ the term "progressive standards."

2 The first fundamental is that *no two things are alike;* the second, that the difficulty of maintaining accuracy increases in geometric ratio with each added accurate dimension on the same piece; the third, that no machine or tool under stress can be accurate; the fourth, that the manufacture of interchangeable parts in quantity is a matter of percentage; and the fifth, that irrespective of the method used, quality is a matter of insistence.

3 Nowhere, either in the arts or crafts, do we find two things alike. Nature herself has never succeeded in making two leaves or two grains of sand the same, as far as man has ever been able to determine. Poe, in one of his essays, states that analysis consists of taking two things exactly alike and looking at them and con-

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tinuing to look at them until we discern that which differentiates the one from the other, and our ability in that direction indicates our power of analysis.

4 The nearest approach to mechanical perfection that we know of is found in the Johansson gages, or was until the advent of the Hoke gages; but within their limits these are by no means interchangeable and the interferometer shows not only a variation in size but a difference in parallelism of the same piece.

5 Securing one very accurate dimension on a piece is a comparatively simple matter. The ease of securing two accurate dimensions, however, depends upon the relation of the second to the first. The figures given in the following table are based upon practice and general impressions, but the author believes them accurate enough to justify their publication for the purpose of showing the high cost of unnecessary accuracy:

No. of Dimensions on One Piece	Probable Number of Perfect Pieces	Per Cent Estimated Increase in Ratio of Cost per Operation
1	100	0
2	90	30
3	50	75
4	15	100
5	5	200
6	0	500

6 In making the foregoing statements the author has in mind automatic and other machines where the close-dimension work is done at one setting and in manufacturing quantities. Some will dispute the figures given in the table and will declare that it is not good practice to attempt finishing pieces in the automatic, but that they should be roughed out there and finished on a shaving lathe or elsewhere. This brings us to our third fundamental and we may ask: why should they not be finished complete when a good automatic must, in the nature of things, be (and is) as accurate as a single-purpose machine *on a single accurate dimension?* It is not accurate on several close ones, however, and accepted practice confirms this statement; and the reason why it is not and cannot be accurate is because of conflicting stresses.

7 The extent to which we may go in accuracy of interchangeability is often determined by the price that can be secured for the

finished product. This is also true where the system of building rather than manufacturing is followed, and where a much better class of labor is necessary.

8 There are many firms the engineering departments of which really believe they are producing an interchangeable product of close dimensions, but their inspection and manufacturing departments could tell a different story. It is not bad workmanship or lax inspection that is responsible for their failure to produce such work, but the — ofttimes — unnecessarily close tolerances specified on unimportant dimensions, or the insistence of close ones on several dimensions of the same piece, and it is well to stop and consider what may happen, say to a piece of apparatus after it has been in service for some time when initially it required the centers of two shafts to be held within one-half thousandth of an inch.

9 Interchangeable manufacture requires both *relative tolerance* and *specific tolerance*. Relative tolerance has to do with its relation to the part to which it assembles and does not necessarily affect the tolerance of the specific dimension. Specific tolerance is that tolerance on a specific dimension required to render a particular part easy to manufacture, or to take care of the wear on tools.

10 Any part increases in cost with each succeeding operation, and the probability of loss should decrease in the ratio of its added value. This result should be obtained, first, by a design having in view its relation to subsequent machining operations, and, second, through the proper sequence of operations relative to their difficulty, and sufficiently divided. This leads us up to the question of registration. Automobile engine builders cast lugs on their cylinders to insure parallelism of bore; adding-machine and phonograph castings sometimes have bosses cast on, to take pressure of milling or drilling operations. It is also true that sometimes we insert a pin in a drilled hole to guard against movement, but we do it only at times and usually as a matter of convenience, whereas it is a matter of necessity; and it will usually cost less to drill special holes or machine special lugs for registration and resetting than to attempt to do the work in fewer complex operations.

11 Nearly as important as registration is the question of clamping. One of the fundamentals laid down was that a machine under stress could not be accurate. This is just as true of a piece being machined, and unless a part is designed with its subsequent machining operations in view; unless it is supported sufficiently

near its pressure centers; unless it has a three-point support with the holding or clamping pieces immediately over them (and in the case of a drill jig, independent of the part that carries the bushings), then that piece cannot be accurate. This is true of drilling always, of milling generally, and of turning sometimes.

12 How tolerances shall be indicated; whether they shall be identical, independent, or overlapping; what the law of probability and what actual trial demonstrate as the probability of overlapping or identical tolerances interfering; the percentage that may be expected at different parts of the tolerance, and the lessons to be learned therefrom are subjects calling for extended treatment by themselves. The same is true of inspection, of machining methods, of analysis of product.

13 The automatic screw machine will, of course, always be with us, for in that we meet ideal manufacturing conditions as nearly as they may be met. But the fact that the single-spindle screw machine persists and is even exclusive in the small-part field, is still further corroboration of the price we must pay for accuracy. The question of accuracy is of course relative, as is the question of rigidity of machine; but it is important in its bearing upon the cheapness of manufacture by determining the number of cuts, retapping, reaming, grinding, etc., that are necessary.

14 Quality does not depend upon the method under which work is done, but it is usual to assume that men working for day wages take more time and consequently turn out better work than those on a piece-work or premium basis. I believe people who reason thus are wrong in their psychology, for the difference between the two methods does not in itself carry any incentive either to do better work or worse work. It is the remuneration received on the premium basis that gives the incentive; and where a premium does not work a wrong by inducing over-exertion, it lessons fatigue by increasing the workmen's interest in his task; and as Taylor says, "Speed is a function of quality."

15 The law of compensation applies to mechanics as well as elsewhere in industry, and when we attempt to work to closer limits at the expense of increased operations, we must pay somehow. This is not to be considered as an argument against such a procedure, but an appeal for common sense in interchangeable manufacture — not to make the work easier, but to reduce the cost — and the firms that are really making a good interchangeable prod-

uct are those that have analyzed all the different conditions and hold the extremely accurate dimensions at a minimum.

16 Neither should it be thought from what has been said that close dimensions may not be necessary or desirable. Some companies require them much closer than do others. It then becomes a question of whether the price received for the finished apparatus is commensurate with the close limits imposed. If not, then it is a matter of increasing the tolerances so as to permit manufacture on a cheaper basis. In other words, the percentage of rejections that can be tolerated must be figured out and kept within that limit. For instance, on an apparatus costing $10.00 for which a liberal price is asked and received, an allowance of 50 cents per apparatus for rejections may not be excessive. If, on the other hand, the price is close, 50 cents may mean the difference between profit and loss. This is a matter of policy to be settled by the administration and not by the shop, although we very often lose sight of this fact.

17 In the author's opinion there are no such things as close tolerances. All are relative and we only court trouble when we try to take too many steps at once. One-half thousandth is only five per cent of ten thousandths, and the chance of securing that accuracy in quantity in one step is about five per cent multiplied by the extra cost. But one-half thousandth is *fifty per cent* of one thousandth and the probabilities are increased in the same ratio, so we may lay it down as a truism that subdivided operations are a function of accuracy.

18 Analyzing our fundamentals, we find that there are three ways in which interchangeable parts may be secured:

a By obtaining a percentage of good ones, with close tolerances

b By giving individual attention to each piece

c By employing liberal allowances.

The first is wasteful and the second is not manufacturing; the third one means liberal unnecessary allowances and close necessary ones, with the operations so divided that each individual working upon the part has but one thing to do. Thus, on a small shaft with six diameters all ground to a 0.0005-in. limit, there should be six roughing and six finishing operations because different wheels may be used; because less skilled men may be employed with less chance of scrap; because the wheel will be in better condition and

will not need dressing so often and the operator will not have to change his sense of proportion, " hog off " material one moment and hardly touch it the next; because the finishing operations may always be done on the most accurate machine; and last, but not least, because as a rule one man can finish more work to close dimensions in four operations than four men can in one operation.

DISCUSSION

CHARLES B. COLE said that it was his experience that difficulties in interchangeable manufacture were sometimes due to the designer who would set plus and minus tolerances the total of which would be greater than the mean tolerance. Another difficulty was in the designing of gages. He had seen a gage a quarter of an inch thick, four and one-half inches long and about three and one-quarter inches wide which contained twenty-one gaging points. On this gage on the smaller dimensions there were limits of 0.0002 in., due to the fact that it was a solid gage with no possibility of salvage or replacement of the particular points which showed signs of wear. Adjustable limit snap gages set to gage at very close limits would frequently be used by an operator and forced over four or five pieces of production, thereby distorting the setting of the gages; and the operator would continue to use this gage without rechecking, so that the results due to distortion through the use of the gage on further parts the variation in the sizes of these parts might run as high as $\frac{1}{32}$ in, from the standard size required.

For interchangeable manufacture great care must be given to gages and tolerances so there will not be any unnecessary loss or cost in production due to setting limits that are too close and are not necessary from the standpoint of good manufacturing. It is in this way that the engineer can be of great help to the production executive. In a great many plants they will attempt to work the limits which are impossible to obtain due to the heat generated in producing these parts which causes a change in the setting of tools and in the gages used to check them. Very careful attention must be paid to the design of any article or product to be manufactured in quantities as it is often possible to simplify them. It is also necessary to design gages and tools with the idea of simplifying

them so they can be readily used by the most inexperienced help. All designs of tools, gages and product must be checked very carefully if interchangeability is to be obtained.

F. O. HOAGLAND said that one fault with interchangeable manufacture was the improper determination of the location surfaces, as, for instance, a piece to contain two holes which are drilled from opposite sides and presumed to be the proper distance apart.

No. 1799

LOCATION AND DISTRIBUTION OF THE CENTRAL-STATION POWER OF THE MIDDLE WEST [1]

By W. L. Abbott, Chicago, Ill.
Member of the Society

THE location and distribution of the central-station power of the Middle West is a subject which has neither the large commercial possibilities of the proposed superpower system of the North Atlantic states nor the spectacular engineering interest of the hydro developments of the Pacific states. The power development of the low, level, rich prairie country, with its wealth of cheap coal, is peculiar to itself, and the distribution is working out its destiny following its natural instincts.

2 Fig. 1, taken from a recent issue of the *Electrical World*, is an ingenious redistribution of the area of the country, designed to show up the various states in their true electrical importance. This map, which represents relative kilowatt-hour production, is doubtless a fairly accurate representation of installed kilowatt capacity, the total for the United States being given as 12,600,000, of which about one-third, or 4,000,000 kw., is hydroelectric.

LOCATION OF THE WATER POWERS OF THE MIDDLE WEST

3 The various potential power streams of the country and the water powers are generally located where they will do the most good — that is, remote from coal mines — the Middle West, in comparison with the rest of the country, not being particularly favored. The states of Ohio, Indiana and Illinois, which are richest in coal, are poorest in water power. Only 18 per cent, or 776,000 kw., of the water power of the country lies in the middle-western

[1] For discussion see paper No. 1800.

Presented at the Spring Meeting, Chicago, Ill., May 23 to 26, 1921, of The American Society of Mechanical Engineers.

states, and the greater part of this is in the states of Michigan, Wisconsin, and Minnesota. Portions of Ohio rise 500 ft. above the Ohio River and above Lake Erie, but the whole state is credited with only 40,000 kw. installed central-station hydroelectric power.

4 In Southern Michigan and Northern Indiana is a land elevation of more than 500 ft. above Lake Michigan, giving the St. Joseph River and its tributaries a fall of 10 ft. to the mile over its entire length. The effect of this fall is to dot the map with water powers all along the river's course. A similar elevation near Detroit

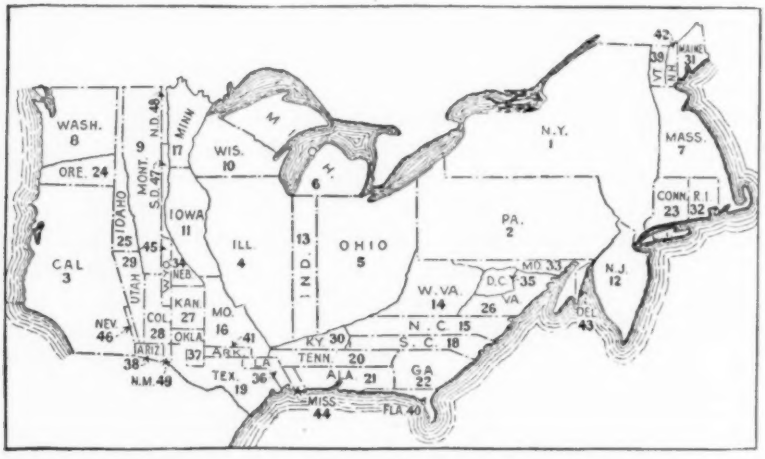


FIG. 1 RELATIVE SIZES OF STATES BASED ON THEIR KILOWATT-HOUR PRODUCTION

has its effect on the Huron River, and a hogback up through the center of the southern peninsula makes good mill streams of the Au Sable River, of the Manistee, Grand and Muskegon.

5 Along the southern shore of Lake Superior the land elevation is, in places, 1000 ft. above the lake, but the territory drained is rather restricted and the rivers are so short that the streams are torrential for a brief interval and almost dry much of the time.

6 The lower Fox River, between Lake Winnebago and Green Bay, Wis., has a drop of 170 ft. in 28 miles, along which over 40,000 kw. of power was installed long ago and fully occupied by local industries. The level plateau of the watershed of the Fox River and the large pondage of Lake Winnebago make the lower Fox River an ideal one for power. The principal discharge for the rivers of

Central Wisconsin is the Mississippi River, which is about 700 ft. above sea level, and as the center of the state abounds in lakes and marshes at elevations of 500 to 1000 ft. above the mouths of the rivers which drain this interior, many important water powers are being developed on these rivers, as at Kilbourn, 6000 kw.; Prairie du Sac, 18,000 kw.; Eau Claire and vicinity, 50,000 kw.; and St. Croix Falls, 18,000 kw.

7 The principal central-station water power of Minnesota is 50,000 kw. in the vicinity of Duluth. The great water power of the Mississippi River that made Minneapolis famous is principally taken up by flour mills and other local industries. About 33,000 kw. of this power is classified as central-station power.

8 The territory comprised in this survey includes and extends from Western Pennsylvania to Eastern Kansas, from Southern Kentucky to the Great Lakes and the Canadian border. It comprises an area of 660,000 square miles and a population of 40,000,000. It all lies in the Mississippi Valley and scarcely one per cent can be rated as mountainous, its broad stretches of prairie and forest lands having in general such slight declivity and absorbent soil that the water run-off, excepting that lost in floods, does not exceed one-fifth of the precipitation, and its placid streams for the greater part of their courses flow to the lakes or sea with a slope of half a foot to the mile.

9 Such a gentle fall of course makes possible relatively little water power, considering the length and breadth of the streams, but as compensation, because these streams have not eroded the soil, the prairie states, with their unrivaled fertility, will some day support a teeming population of hundreds of millions, whose increasing clamor for power must be satisfied principally, as now, by coal drawn from their own great stores.

10 Water power must be developed at the waterfalls, and until long-distance transmission was developed that power had to be used where it was generated; and because much of such power was located in rough, inaccessible country, having no other features of commercial importance, such rivers were allowed to roll on in solitude "and hear no sound save their own dashing." Now electric transmission has changed all of this, and whereas once factories went to the power, now power goes to the factories, and with a widening effective range of transmission lines and higher cost of coal, more and more of these wild waters are being broken to harness and few of those remaining are escaping serious consideration.

THE QUESTION OF POWER PLANTS AT MINE MOUTH

11 The example of hydroelectric power transmission naturally suggests that coal be burned at the mine to develop steam-electric power for transmission to remote markets, thereby saving the expense and uncertainties of rail shipment. The argument seems plausible, and yet the author is not aware that such a thing is being done anywhere in a large way. Indeed, it is doubtful if in any but the most favorable cases power plants at the mine mouth with a transmission line can successfully compete with power plants located in the remote centers where the power is to be used, the coal necessary for these plants being shipped by rail.

12 It must be admitted that during the past few years the railroads have done much toward popularizing electric transmission, but rates have not quite reached the point where, even under most favorable conditions in the Middle West, a power company would be warranted in maintaining a power plant at the mine, a stand-by plant in the city, and a transmission line between. Conditions, moreover, are seldom "most favorable," as with us the large quantity of flowing water necessary for condensing in a plant of upward of 50,000 kw. is seldom found at the mines. This renders it necessary to ship the coal to a power house located on a nearby river and to transmit electrically from there to the place where the power is to be used, but right here the railroads that have tried so hard to popularize all-electric transmission have unwittingly intervened with an awkward freight-schedule system, which renders a part-rail and part-line transmission impossible. This has happened because they did not know there was any system to their schedules, and while the author cannot say that these schedules were prepared according to any rule, he has nevertheless found a key that will fit the average of several taken at random. This key indicates that a coal freight schedule is composed of a basic charge of $1.00 a ton and a hauling charge of four-tenths of a cent per ton-mile. Let us see how this affects a rail and line transmission.

13 From the Springfield coal district to Chicago is 192 miles. This distance, at 0.4 cent per ton-mile, amounts to 77 cents; adding the basic $1.00 makes $1.77. The actual rate, $1.83, is a little less that 1 cent per ton-mile.

14 There being no dependable supply of condensing water near Springfield, it has been suggested that the coal destined for Chicago power be shipped to Peoria, where there is water in

abundance, used there in power houses, and the electric energy sent to Chicago via a high-tension line.

15 From Springfield to Peoria is 62 miles, which at 0.4 cent is 25 cents; adding the basic $1.00 gives $1.25. The actual rate is $1.17, — a little less than 2 cents per ton-mile, or about double the ton-mile rate from Springfield to Chicago.

16 Obviously, the Springfield-Chicago all-rail shipments need have no fear of competition via the Springfield-Peoria-Chicago rail and line route.

LOCATION OF POWER STATIONS AND DISTRIBUTION LINES IN THE MID-WEST TERRITORY

17 Fig. 2 is a map showing the distribution lines of the territory discussed and also showing by dots and circles the centers

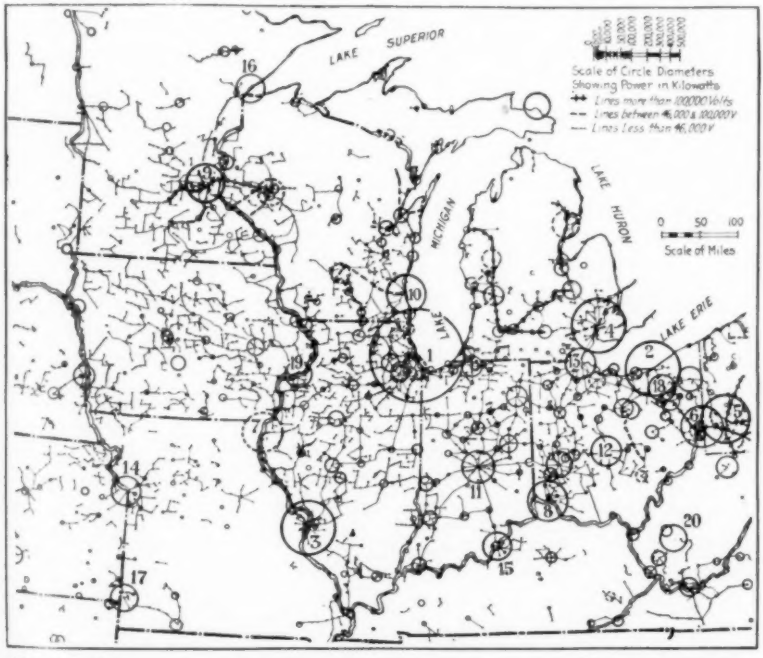


FIG. 2 CENTRAL-STATION POWER DISTRIBUTION OF THE MIDDLE-WESTERN STATES

1—Chicago, Ill.	6—Windsor, W. Va.	11—Indianapolis, Ind.	16—Duluth, Minn.
2—Cleveland, Ohio	7—Keokuk, Iowa	12—Columbus, Ohio	17—Joplin, Mo.
3—St. Louis, Mo.	8—Cincinnati, Ohio	13—Toledo, Ohio	18—Akron, Ohio
4—Detroit, Mich.	9—St. Paul, Minn.	14—Kansas City, Mo.	19—Moline, Ill.
5—Pittsburgh, Pa.	10—Milwaukee, Wis.	15—Louisville, Ky.	20—Cabin Creek Jct., W. Va.

where the power is generated and used. The smallest circle shown represents 500 kw.; smaller amounts of power are indicated by dots. Central power stations of less than 500 kw. and not connected up to another center do not appear on the map. Broken circles indicate a power station, the greater part of whose output is included in the area of some solid circle. The developed central-station power of the Middle West is 4,500,000 kw., of which all but 18 per cent is from steam power.

18 The large circle, indicating 500,000 kw., for the central stations of Chicago and vicinity, is the dominating feature; following in order are Cleveland, St. Louis and Detroit, each with a little more than 200,000 kw. The 210,000 kw. of St. Louis is made up in part by its own Ashley Street Station and in part by the 100,000 kw. from the Keokuk dam, shown as circle 7. One significant feature of the important high-voltage transmission lines indicated in the figure is that such lines are used almost entirely to connect water powers with centers where their power is used, the exception to this being in the vicinity of Pittsburgh, where mine-pit cars dump their coal into the hoppers of boiler rooms located on river banks, and in Detroit, where a small amount of power is distributed at 46,000 volts; also a short 66,000-volt line owned by the Doherty interests in Central Ohio.

19 The line over which the greatest amount of power is transmitted connects the Stone & Webster Keokuk power with St. Louis at 110,000 volts. Probably the system next in importance is that of Hodenpyl & Hardy in Michigan, connecting up water powers on the Au Sable, Manistee and Grand with a line of 140,000 volts — the highest tension in the Middle West — with some distribution at 70,000 volts.

20 In Wisconsin the important hydroelectric developments on the lower Wisconsin River at Kilbourn and at Prairie du Sac are connected by 66,000-volt lines to Milwaukee and to important manufacturing centers to the south, which are included in the Middle West system. The water powers of the Chippewa River and its tributaries have been developed largely by the Brewer interests. Much of this power goes to St. Paul and Minneapolis over a 120,000-volt line, and to other Mississippi River cities farther south. The distribution to the south and west from St. Paul and Minneapolis is over 66,000-volt lines and 110,000-volt lines owned by the Byllesby Company.

21 A study of the five principal states of the group represented

in Fig. 2 indicates that when power is to be generated by steam it can best be done by central stations in the locality where it is to be used. For example, Chicago, with its large and efficient power houses, is surrounded by power houses of the Public Service Company of Northern Illinois and by those of the Middle West Utilities Company, with all of which it exchanges power, but which nevertheless generally supply the power needed in their immediate respective vicinities. Chicago is connected with Keokuk and Keokuk with Indianapolis and New Albany, and yet the power sent over these lines from its large power houses is small in comparison with that generated and distributed locally along the lines.

22 Formerly each small country town requiring 50 kw. or more had its own central station, using 20 lb. of coal per kw-hr. As there was no money in that, there came about consolidations of these isolated central stations into small groups, all supplied from the best-located plant of the group. It was found that with an improved load factor and larger units the coal consumption was by this means reduced to 15 lb. per kw-hr. The next step was to connect these groups into larger ones of 5000 kw. capacity, favorably located as regards condensing water and rail connections, having condensing turbines and capable of delivering a unit of energy for 4 lb. of coal. A plant of this size can reach out its arms and gather in all of the power within a radius of thirty miles or so. This is about the most efficient combination of station power and transmission lines, unless it happens that the power demands of the city in which the plant is located are such that the use of generating units up to 15,000 kw. is practicable, in which case a further material improvement in efficiency is secured and warranting the construction of longer lines to reach more remote groups of power, these large plants being interconnected over lines of sufficient capacity to permit them to be mutually helpful.

23 Such is the plan which is naturally developing, and although its progress must be as slow as the process of consolidation of the properties, it appears that the problem of cheaper power production is working out its own solution along conservative lines and that in time the present systems will merge into larger ones, with one general scheme for generation and distribution, so that areas equal to a state in extent and overlapping state lines shall be under one management.

24 What, then, about home rule? Many years ago, when "tariff" was the football of national politics, a candidate for the

presidency, upon being asked to declare his position on this burning issue, replied that the tariff was a local question and had no place in a presidential campaign. It is by such reasoning that the interests of a utility, stretching for hundreds of miles across the country and supplying power to hundreds of scattered or contiguous communities and to thousands of industries, becomes a local question and properly subject to local regulation as a foreign monopoly by each of the various cities, towns and villages in which it renders service.

25 Fortunately it is becoming generally recognized that a state-wide industry of this character requires state-wide regulation and that for the state to abdicate its jurisdiction, cut it up into sections and parcel out the pieces to the police powers of the various localities in which the utility operates, subjecting it to the vicissitudes of local politics cannot help that utility to develop and to most economically and efficiently serve those same communities.

No. 1800

FUTURE POWER DEVELOPMENT IN THE MIDDLE WEST

By C. W. Place,[1] Chicago, Ill.
Non-Member

In this paper the author outlines a system in which it is proposed that the fairly efficient steam power plants in the 100-odd cities of over 25,000 population in the 14 middle-western states be interconnected, not by heavy high-voltage lines, but by lines running from each which will pick up the small-town and village load to the point where its next larger neighbor will take its share; hydroelectric plants are to be installed on the streams near the towns and villages, each with a small pondage; the steam stations are to carry the steady continuous load above the maximum stream flow, the hydroelectric plants automatically coming on to carry the peaks. On the off-peak period the hydroelectric plants will restore the pondage and carry local load. The quickness with which the latter can get on the line (12 to 30 seconds) will enable the steam stations always to work at maximum efficiency. The advantages of the system are pointed out in the paper, as well as the procedure necessary to bring about its development.

UP TO a certain point in the growth of a country or under abnormal conditions, the growth and development are the important things: the means to the end or the cost are not of great consequence. Economy and the conservation of natural resources cannot be allowed to interfere with this early expansion, but there is nevertheless some stage of development and concentration of population where early methods must be changed.

PRESENT SOURCES OF POWER IN THE MIDDLE WEST

2 At present the sources of power for the supply of light, heat and energy for the mechanical operations necessary for the life and continued development of the Middle West are coal-, oil-, and gas-fired steam equipment, internal-combustion and heat engines,

[1] General Electric Company.

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and water-driven equipment. Wind and solar-heat sources need not be considered.

3 Fig. 1 is a map of the Middle West showing the coal fields it draws from, as well as the rivers and larger streams. Table 1 gives the coal production of this region according to Government

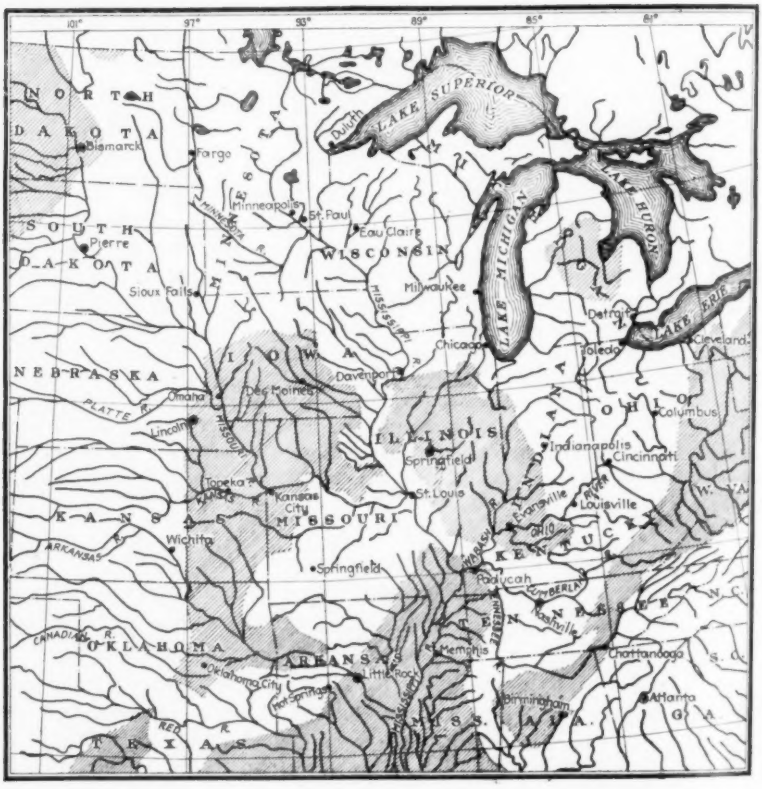


FIG. 1 MAP OF THE MIDDLE WEST, SHOWING ITS RIVERS AND LARGER STREAMS AND THE LOCATION OF ITS COAL FIELDS (SHADED AREAS)

estimates for the period 1917–1920, and Table 2 data on the potential and developed water of the fourteen states it includes. Table 3 shows the oil production of the United States in 1920, that of the Middle West being 15 per cent of the total.

4 It will be noticed in Fig. 1 that the coal fields are not located conveniently as regards water supply for condensing purposes and are not particularly convenient to the load concentrations as at

TABLE 1 GOVERNMENT ESTIMATE OF BITUMINOUS COAL PRODUCTION IN 1000 TONS

	1917	Per cent[1]	1918	Per cent	1919	Per cent	1920	Per cent
Total U. S.	551,790	100	579,385	100	458,063	100	556,563	100
Arkansas	2,143	1.0	2,227	1.0	1,680	1.0	2,310	1.0
Illinois	86,199	41.6	89,291	40.0	64,600	38.5	90,050	40.6
Indiana	26,539	12.8	30,678	13.7	20,500	12.2	30,420	13.7
Iowa	8,965	4.3	8,192	3.6	6,300	3.7	9,170	4.1
Kansas	7,185	3.5	7,562	3.4	5,750	3.4	6,700	3.0
Kentucky	27,807	13.5	31,612	14.2	28,500	17.0	31,000	14.0
Michigan	1,374	0.6	1,464	0.6	930	0.5	1,440	0.6
Minnesota	.....	.....	.....	.....	.....	.....	.....	.....
Missouri	5,670	2.7	5,667	2.5	4,060	2.4	5,750	2.6
Nebraska	.....	.....	.....	.....	.....	.....	.....	.....
N. Dakota	790	0.4	719	0.3	750	0.4	770	0.3
Ohio	40,748	19.6	45,812	20.5	35,050	20.8	45,000	20.2
S. Dakota	.....	.....	.....	.....	.....	.....	.....	.....
Wisconsin	.....	.....	.....	.....	.....	.....	.....	.....
Total, 14 states	207,520	37.6	223,224	38.6	168,120	36.7	222,610	40.0

[1] Percentage opposite each state is percentage of production of the 14 states listed. 207,520,000 tons of 11500-B.t.u. coal will produce 249,000,000,000 kw-hr. of energy at present efficiency, or at the rate of 28,000,000 kw. continuously. The total output of electrical energy in 1919 in U. S. was 3,215,552,000 kw-hr.

TABLE 2 WATER POWER IN THE MIDDLE WEST

(Extract from *Electrical World* Estimate, Jan. 1, 1920.)

State	Potential Maximum	Developed		Undeveloped Maximum	Growth of Industrial Hp. req. in 1930 in Excess of Present Station Rating
		Hp.	Per cent of Max.		
Arkansas	75,000	13,400	17.9	61,600	7,000
Illinois	410,000	191,000	46.7	219,000	1,302,000
Indiana	140,000	123,000	130.0	.....	558,000
Iowa	450,000	423,000	94.2	27,000	78,000
Kansas	320,000	105,000	32.8	215,000	7,000
Kentucky	250,000	61,400	24.6	188,600	62,000
Michigan	350,000	357,000	102.0	.....	1,535,000
Minnesota	600,000	233,000	38.8	367,000	261,000
Missouri	200,000	39,500	19.8	160,500	49,000
Nebraska	280,000	65,200	23.3	214,800	.....
N. Dakota	150,000	1,000	0.7	149,000	10,000
Ohio	215,000	200,000	93.0	15,000	1,150,000
S. Dakota	90,000	11,900	13.2	78,100	.....
Wisconsin	1,150,000	237,000	20.8	913,000	298,000
	4,680,000	2,121,400		2,608,600	5,317,000

Thirteen applications for Federal permits were made up to April 5, 1921, in the Middle West, and 205 applications in the entire United States and Alaska.

TABLE 3 GOVERNMENT ESTIMATE OF CRUDE-OIL PRODUCTION IN 1920

	1000 Bbl.	Per Cent Total of 14 States
Total United States	443,402	....
Arkansas	.....	....
Illinois	10,772	16.2
Indiana	932	1.3
Iowa	.....	....
Kansas	38,501	58.2
Kentucky	8,679	13.1
Michigan	.....	....
Minnesota	.....	....
Missouri	.....	....
Nebraska	.....	....
N. Dakota	.....	....
Ohio	7,412	11.2
S. Dakota	.....	....
Wisconsin	.....	....
Total for 14 states	66,296	15.0

During 1920 about thirteen million barrels of crude oil were used by public utilities for power production. Of this California and Texas used 8,500,000 bbl. and Missouri and Kansas the greater share of the remainder. Crude oil is therefore of relatively little importance as regards the total power requirements.

present arranged. A recent news report indicates that the Federal investigators on power conditions of the Atlantic Coast states have concluded that the solution of the problem does not lie in enormous production at the mines on account of condensing-water conditions. On the basis of our conditions and economical production, every 100,000 kw. produced in a day would probably require 380 million gallons of water, or a flow corresponding to 588 sec-ft., for condensing purposes. It would therefore seem that the demand for power in the Middle West cannot be taken care of at the mines as these are not situated near streams of the proper size. The rivers and streams are in better relation to the load, as the cities and towns have grown up along these streams.

POWER POTENTIALITIES OF THE RIVERS AND STREAMS

5 The production of power thus far by steam has been practically always at the load concentration. A start has been made transmitting from steam stations to less heavy load centers. A moving of load to cheap power has been successful in a number of cases, but this has caused unnatural conditions in other respects and has not necessarily been for the greatest good of the country

as a whole. We have no great sources of hydraulic power conveniently available which can be profitably developed at present money rates and material costs. We must therefore lay our plans on our real conditions and not on some imaginary combination as regards power supply or regrouping the load arrangement. Large blocks of power have been developed at centers of load by large steam stations. These large centers are, however, not close together but each has its tributary country with low load density. This territory is made up of towns and villages surrounded by farming country drained by innumerable streams.

6 Most of these streams have comparatively low banks, have bottom lands which are good farming land and have extreme flood and low-water periods. As one travels along these streams any number of small dams may be seen that are either partly washed out or are still serving to drive some mill and making a more or less efficient use of the water going down the stream. From his observations the author would say that these streams would be able to supply the necessary power for all the towns and villages near them of 25,000 population or less; that during a good part of the year they could assist even in these towns during the peak if properly connected and manipulated. In these fourteen central states there are but 102 cities of over 25,000 people, and cities of this size have already established quite efficient steam plants.

PROPOSED INTERCONNECTED SYSTEM OF CITY STEAM PLANTS AND VILLAGE HYDROELECTRIC PLANTS

7 Suppose now that the power houses of all these 102 cities were connected together, not by heavy high-voltage lines, but by lines running from each which would pick up the small-town and village load up to the point where its next large neighbor would take its share, and that the streams near these towns and villages had hydroelectric stations of a size proportionate to the size of their respective streams. These 102 steam stations must work on a fair load to get their best efficiency both from a steam-economy and a maintenance basis, and when the units are operating at good efficiency a sudden increase in load cannot be handled properly until other units are brought on and steam conditions have been restored. During this period of adjustment the voltage goes down and the frequency droops, and would droop more if the voltage were brought back to normal.

8 Due to these conditions extra machines are now carried either under part load, or ready to put on at once. Suppose the interconnection mentioned above has now been made, and a goodly number of these hydraulic stations installed. These hydraulic stations are large enough to take a little more than the average stream flow, i.e., are overdeveloped. Each has a small pondage and is automatically controlled so that it will come on on a drop of frequency of the system and in sequence with other plants, depending on the proportion of its working head available at the instant. That is, the plant with the best water conditions will get on the line the quickest. It will drop off the line on a decrease of its water supply or a dropping off of load. The steam stations are operating efficiently and have a sudden increase in load.

9 These various hydraulic stations are operating, using up the natural stream flow and sending but a very small proportion of their load toward the city. With the droop in frequency the several water-wheel generators installed for the flow above the average begin to come in and to draw on the ponds. Enough come in to bring the frequency back to normal. In the meantime the steam stations are back to normal. The operators have not let the voltage stay down to save the speed, but have restored it at once and the service has been maintained. With this automatically started reserve there is an opportunity to decide whether the load is going to continue, and if so, to get on extra boilers and machines. It is known that the hydraulic helpers are going to begin to drop out as they exhaust their pondage, so the steam operation can be governed accordingly.

10 Again, suppose that each of these streams had the low swampy ground and the willow thickets made into ponds even where there was insufficient head for power development, which could release when the ponds with hydroelectric stations were drawn down by this sudden drain, and close when the station pond was restored. This would mean perfect flood control for normal rises, and, by holding out the steam stations, very good control during extreme floods.

11 These smaller streams lead to the larger rivers and with an approach at flood control on the tributaries there would be a greater number of locations for hydroelectric development on a larger scale along these larger rivers, each to be controlled in the same way. Each development would help the flood conditions below and make feasible developments not now economically practical.

12 Very closely resembling this hypothetical case is the arrangement for taking care of the peak load of New York City suggested by F. O. Blackwell at a meeting of the New York Section of the American Society of Civil Engineers, February 9, 1921. Mr. Blackwell stated that water if pumped up to an area on the Palisades during the peak, would develop the peak power more economically than it could be done by any other method of reserve. He further stated that this is being done in several installations in Switzerland, the water being pumped into reservoirs when the load is light. Prof. R. A. Fessenden in April published an article suggesting that such storage be made underground, the hole being dug near the load.

13 Our conditions do not perhaps lend themselves to this arrangement as readily as the combination mentioned previously. There are, however, cases where these schemes would probably work out. Along certain parts of the Mississippi River there are high bluffs and in some places the land is not particularly good on top of the bluff. A transmission line along the bluff might obtain the reserve capacity in this manner if at the same time the pondage could act as a reserve reservoir for city filter beds or some such purpose. There are high valleys in the Ozarks where there is no natural large water supply and where the land is of little value which could act as such a reserve if properly arranged and if in conjunction with the natural water-power supply in the same region. This might even make some of the powers with greatest water fluctuations economically feasible.

14 As regards Chicago, there is quite a large reserve at the east in Lake Michigan. On the way to Lockport there are large areas of worked-out stone pits and ground where it might be possible to get short-time storage so that the drainage canal could be drawn upon for such peak service without unduly drawing down the lake level. Excess capacity would have to be installed at Lockport.

15 The State deep-waterway project could be overdeveloped and operate along this line if properly connected and operated.

16 All this sounds highly imaginary, but it is being done on a small scale in Saline and Seward Counties, Nebraska, along the Big Blue River, with manually operated and automatic stations, except that as yet there is no steam reserve. This system is being extended by the addition of more stations, and from present indications the entire river will soon have increasingly larger stations installed, the dam of each being at the point where the ordinary

backwater from the dam farther down the river runs out. Other systems are also doing something along this line.

ADVANTAGES OF THE PROPOSED SYSTEM

17 Suppose this idea were carried to its ultimate conclusion. The steam station would then carry the steady continuous load above the minimum stream flow, and the hydraulic equipment would carry the peaks, and on off-peak periods restore the pondage and carry local load. The quickness with which the hydraulic stations can get on the line (12 to 30 seconds) will allow the steam stations to work always at maximum efficiency.

18 All of these developments must be made at a cost that will show a reasonable profit. In order to arrive at the correct profit, however, all the advantages must be weighed. A small automatic station tied to the line has a stabilizing effect on voltage which should be worth something. It should receive a credit equivalent to the transmission losses and another on the basis of the investment in coal storage of the steam station which it relieves; it should also receive some credit due to absence of labor trouble.

19 A multitude of these small stations will, by means of legitimate interconnections, do more actual work than the same installed capacity in one station. The small station made automatic can usually be installed on a much more satisfactory basis than the large plant. It can usually be financed locally, and a good share of the money will remain near the site. A large amount of land will not be taken out of production, so there will not be a large overhead, and the land taken will be of small value. If the development is forced beyond a certain point under middle-western conditions, this land becomes the most valuable corn land in that section, i.e., the second bottom land. The dam which before was an easy proposition on a more or less soft bottom, now becomes a difficult one, and the development will not show a profit. It is therefore necessary to multiply the number of stations instead of increasing the size if we wish to get total capacity. If these smaller stations are first installed it would seem that the larger streams could be used if developed on the same general plan and money made by the development.

20 Fig. 2 shows the daily load curve of a city having steam- and water-driven power supply, and having lighting, power and railway load served. If the water could be called to carry the sudden

rise and the extreme of the peak it can be seen what an advantage it would be. It is evident that this could not be done with a single source of water power with any storage available in the river valley, but only with a number of such reserves acting for a territory rather than for one city.

21 What, now, are the things that we should do? In the opinion of the author —

1 The stream conditions and their possibilities should be

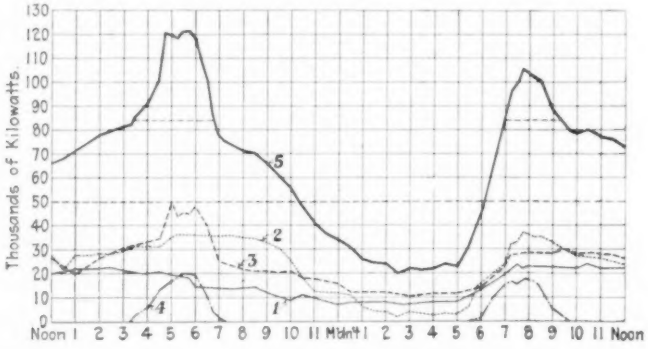


FIG. 2 DAILY LOAD CURVE OF A CITY HAVING STEAM- AND WATER-DRIVEN POWER SUPPLY FOR LIGHTING, POWER AND RAILWAY SERVICE

Curves	Kw. Peak	Kw-hr.	Load Factor, per cent
1—25-cycle steam generators.....	24,000	362,150	62.8
2—60-cycle steam generators.....	36,600	552,490	62.9
3—25-cycle hydroelectric generators..	50,000	528,000	44.0
4—Railway Co. steam generators...	20,180	79,610	16.4
5—Total.....	121,500	1,522,250	52.5

studied with the above-described or some similar method of utilization in view.

2 Public opinion should be guided in the proper direction so that it will not obstruct such development.

3 Rivers which are technically navigable but are not really so should either be released or made navigable and the flow used for power development, and the engineer's influence should be used to bring this about.

4 In the permanent-highway-improvement campaign, influence should be used to have these permanent highways diverted from sites capable of future development or their natural ponds.

5 The relative importance of railway transportation and

power development may change, so that any readjustments of railway roadbeds up river valleys should be along lines that will not interfere with the possible stream development. It may easily prove that the railway may be the first and most important benefactor in its electrification. Public sentiment might very easily assist in this.

6 Existing power sites in the form of mill dams should be developed and tied into the nearest transmission, and the possibilities in this direction should be investigated and taken advantage of in every locality.

7 The investing public should be instructed regarding developments in their vicinity. They should be shown that our water-power possibilities are not gold mines for the owners and are of little value if not tied in with existing systems; but that when they are so tied in they increase the value of the system and make its securities much better from the investment standpoint.

DISCUSSION[1]

P. F. WALKER. This paper is suggestive and most interesting to the writer because of his work on the research committee of the Mid-Continent Section. It represents work similar, also, to investigations under way by committees of the Kansas Engineering Society and by members of the staff of the Engineering Experiment Station of the University of Kansas. The fact that the system already employing some of the methods advocated by the author is in Kansas lends further interest.

Automatic remote control of hydraulic power stations has brought this new possibility into being. It means that many small developments may be made possible on streams where flow is small during a large portion of the year, and made to serve the useful purpose of reducing the necessary capacity of steam stations by carrying the short-period peak load each day on pondage water. An economy not mentioned by the author may be possible often where there are several steam stations connected with the water stations. During times when the flow of one or more streams is in excess of the water plant capacities, and this would be the case

[1] Refers also to paper No. 1799.

during many consecutive weeks of the year, one or more steam plants could be shut down and the water plants made to carry the regular load in corresponding measure.

At the present time in Kansas there are in operation twenty-three water stations with a total capacity of 9572 hp. Fourteen of these are in public utility service. There are several more undeveloped projects known to the present writer, some of which are so placed that they would adapt conveniently to extensions of existing transmission systems. Many of the present plants could be increased in capacity if they were to be operated on the over-developed plan for peak-load service. This means that much more could be realized from water power in this state than has been generally supposed, were there carried into effect the proposition of connecting up all of the transmission systems. Doubtless the same is true in other states.

The importance of water resources is just beginning to be realized in many of the states. It will have a determining influence on general industrial development. Publicity on such matters as these, after careful investigation, forms one of the services which the engineer may render to the public.

G. W. HAMILTON.[1] In the situation at Chicago where approximately two billion kw-hr. are generated per year on a basis of two lb. of fuel per kw-hr. and the freight rate of a dollar per ton of fuel there would be an annual saving of two million dollars if power could be generated at the mine and the cost of transportation by freight eliminated.

Many of the points brought out by Mr. Place have been developed for a number of years by The Middle West Utility Company, where the most efficient steam stations are being operated and small water powers are fed into the transmission line wherever feasible.

W. E. BRYAN.[2] The writer is connected with a company which uses approximately 50 per cent of the power from Keokuk and he has been particularly interested therefore in the paper by Mr. Place. Keokuk is 142 miles from St. Louis and the local light and power company has always felt that while the greatest possible use should be made of Keokuk water power certain stand-by machinery must necessarily be provided at St. Louis to insure continuity of service. The result is that there is more steam generating

[1] Middle West Utilities Co., Chicago, Ill.

[2] United Railways, St. Louis, Mo.

equipment in St. Louis than is necessary to supply the usual demand. There are two lines between Keokuk and St. Louis, either one of which is capable of carrying the total St. Louis load, and at no time have both of these lines been out of service. Interruptions have been very infrequent and the few that have occurred have been due to slight errors, principally in switching operations at Keokuk. However, on one or two occasions, in severe winter weather, one line has been out of service for twelve or fifteen hours and shortly after being restored to service the other line went out. The writer recalls one or two such cases, in which the St. Louis utilities were ready to utilize the complete capacity of their equipment upon a few minutes' notice if conditions should make it necessary.

This is a condition which must be recognized. It might be possible to increase the equipment at Keokuk, but in any event there must be maintained at St. Louis a greater capacity than is needed from day to day in order to provide for emergencies. If the additional power which Mr. Place speaks of were carried by Keokuk and not by St. Louis steam, it would result in a saving in operation, perhaps, but the fixed charge would continue, not only on the steam plant as it now exists but on the additional water-power plant as well.

GEORGE A. ORROK. Thunderstorms are prevalent everywhere in this country. In long-distance transmission service of hydro-electric power to a large city located in a region where thunder-storms are numerous there will be times when this power will be out of service. We have not yet arrived at the point where it is possible to build an electric transmission line which is proof against a direct stroke of lightning.

The writer recalls a situation in which three lines built over two separate routes were completely out of service at one time due to a thunderstorm. Three or four hours elapsed before hydroelectric service was restored. This means that practically a full steam reserve is necessary if dependence is placed on long-distance transmission lines and continuity of service is essential.

Mr. Place brought up another point which the writer may call a water storage battery. This is the use of spare hydraulic power at the time of low load to pump water to an elevated reservoir to be used to generate power at the time of peak load. There are seven or eight installations of this nature in Europe the largest of which is about 5500 kw. The storage pond, however, is the crux of the situation.

If there is a natural storage pond at the top of a hill this hydraulic power can be used with a short length of pipe line at a reasonable rate. If it is necessary to create such a reservoir it will be found that the cost of installation will exceed any economy obtained by it even though the power costs nothing.

JOHN A. STEVENS. In England the writer found that the entire British Isles were divided into thirteen zones similar to the suggestion made by Mr. Abbott. The less economical station must purchase from the more economical station, but if for any reason the better station cannot be continued or extended to take on additional load for the time being the less economical station is permitted to install apparatus to tide over the emergency until the better station can take care of the load at which time the poorer station must suspend operation. An electricity commission made up of engineers has charge of this relation with power plants.

In Italy the situation is even more interesting. If there is power for sale at certain periods of the year in the Alpine section and if a purchaser in the plains section can make use of this power he must buy it rather than make it himself. If the basic cost of power in the Alpine section should be two cents a kw-hr. while it could be produced at the cost of one cent an hour by the purchaser in the plains a charge is permitted to protect his investment at this rate. In other words the investment is protected by the government.

SAMUEL INSULL.[1] The development of interconnection of transmission systems is going forward on economic lines. In the state of Illinois there is a committee devoting its time to this subject. This movement was started by the Commonwealth Edison Company and similar movements are going forward in Indiana, Wisconsin and as a result are extending to Minneapolis and southward from St. Louis and Louisville because these regions are served by a few large groups of operators and because there is practically a continuous system of distribution over them. This same policy is being followed in the Rocky Mountains and has been followed in southern California, in Texas and in Oklahoma.

In some locations where different combinations of capital operate in the same territory transmission lines are practically joint bus lines. In the writer's opinion it is more desirable so far as the Middle West is concerned that development should continue along these lines.

The writer agrees that a hundred- or a hundred-and-fifty-mile

[1] Commonwealth Edison Co., Chicago, Ill.

transmission line cannot be depended on to give service to a big central population. Chicago, like St. Louis, has been educated to continuous service because its transmission system is mainly underground. The territory around Chicago has been educated to an overhead system with its accompanying troubles and these people do not wish to pay the price that the introduction of an underground system would demand.

With all the various distribution systems interconnected a situation will be created which will result in a system of transmission lines not unlike our system of trunk-line transportation systems. In the pioneer work of establishing power stations and a general distribution system the engineers of this country can lay the basis of ultimate electric transportation and ultimately cheaper transportation. If those interested in transportation will confine themselves to the management of transportation and allow mechanical and electrical engineers to conduct the manufacture and distribution of energy there will gradually be created throughout the country conditions which will enable energy to be produced at the lowest possible cost. This energy will be fed into a general network of distribution systems with the smaller stations used at times of peak load. With the diversity of demand which exists in city and country there will be provided a load factor which will not fall far short of continuous use.

Notwithstanding the scheme of regulation which Mr. Stevens has spoken of as existing in Europe the writer believes that there is no place where there is greater freedom of action to the engineer and to the capitalist and to the commercial man than in the United States. The writer is familiar with the situation in England having followed it for many years. There are sixty-one separate undertakings in the County of London. Before the war their average cost was greater than the average selling price in the city of Chicago. There is a population of seven and a half million of people in the County of London and at that time there was a population in Chicago of two and a half million. The production of energy in Chicago with this population was greater than in London and in spite of the new law the best site for a power station for the County of London cannot be used because it is not permitted by government authorities. There are no such restrictions here.

There are certain restrictions, however, which should be made here. No company should be allowed to manufacture energy that can be bought at lower figures. Public utility operators have ac-

cepted governmental regulation for years. Regulations are now going further in the direction of protecting the public. Mechanical and electrical engineers and the men engaged in the business of producing energy should not be allowed to commit economic errors from year to year and our state utility commission instead of being deprived of powers to regulate the whole industry should have powers to regulate it more drastically in the direction of the policy which with mechanical development is bound to succeed because it is economically correct, and that policy is a distribution of energy not confined to a community or to the state but interstate in character, just as railroad transportation or telephone communication.

C. W. PLACE. The interconnection of large power stations is certain to happen in the near future. The writer can see no difference as regards reliability between power from a hydroelectric station feeding into such an interconnected network, and power from a steam station feeding into such a network. Lightning and line troubles will not differentiate between the two sources of power. No one would advocate depending upon one line or one source of power for an important section, but rather the operation of a zone and a large number of sources as mentioned in the paper. The writer has advocated a complete network of lines connecting the extremes of capacity from the large city station down to the small hydroelectric station. This means that power will easily flow to points in the network where there is heavy load.

No. 1801

ANNUAL REPORT OF THE COUNCIL

THE Council is confident that the year 1921 will show in later years quite a marked progress in the whole engineering profession, both as to actual results obtained and also appreciation of those results on the part of the community.

The Society has taken a conspicuous part in the year's developments, and its members have performed this work as a privilege and with due assumption of the obligation of professional men in their relations with the public. In the nation this work has been through the instrumentality of the Federated American Engineering Societies, and locally it has been through the Sections of our Society both individually and in coöperation with other local engineering societies. The participation in these activities by the Society and its committees and members generally, has resulted in the maintenance of a particularly fine feeling and spirit throughout the Society.

SPECIFIC WORK OF THE COUNCIL — THE NEW CONSTITUTION

Inasmuch as the redrafting of its Constitution marks an epoch in the history of any organization, so it should have special mention in any account of the work of this Society.

The year saw the formulation of a proposed new and more simplified Constitution, decided upon at the March meeting of the Council in Boston, Mass.

The first rules of the Society were adopted in the year of organization, 1880, and were in force four years, when they received general revision. This Constitution served the Society for ten years, and in December 1903, under the direction of the Council, a third Constitution and By-Laws were formulated by a special committee and were adopted by the Society and put into effect. This is the Constitution under which we are still operating. Therefore, the present Constitution is eighteen years old.

During the last two or three years the Society has expanded very rapidly and the Council has found it necessary, on recommendations of standing and special committees, to make pronouncements of our policies and of our changes in policies necessitated by this expansion. As far as possible, changes in the Constitution and By-

Laws have been made in keeping with the adoption of these new policies, but as would be expected the present Constitution and By-Laws, by virtue of piecemeal amendment and in many cases too great detail, needed redrafting and simplification.

Under the circumstances the Council agreed that it would be wiser to rewrite the existing document entirely, and accordingly placed the matter in the hands of the Standing Committee on Constitution and By-Laws. A new Constitution was drafted by the Committee and was referred by the Council to the membership in session at the Spring Meeting in Chicago. Here it was discussed in full and, in due procedure, mailed to the entire membership. It was presented at the Annual Meeting, 1921, for second reading, and was again fully discussed both at the Business Meeting and at the Conference of Local Sections Delegates.

Many suggestions of value were received and all of them are being considered by the Committee on Constitution and By-Laws, consulting with the Council and the Local Sections' officers, in preparation for a new presentation of the Constitution at the Spring Meeting in Atlanta, Ga., in May 1922. In accordance with the existing Constitution, this new presentation will again become a first reading of amendments to the Constitution.

FINANCES FOR THE YEAR

The Finance Committee was able to report that the total gross receipts of the Society from all sources for the fiscal year were $497,401.18 and the gross expenditures chargeable to current receipts, $479,419.10, which left an excess of receipts over expenditures of $17,982.08.

BALANCE SHEET

SEPTEMBER 30, 1921

Assets

Society's one-quarter interest in the Building, Land, and Real Estate Equipment (25 to 33 West 39th Street)		$486,792.79
Library Books	$13,000.00	
Furniture and Fixtures	5,000.00	
		18,000.00
Stores, Including Plates and finished Publications		43,939.90
Engineering Index		10,000.00
Trust Fund Investment:		
Liberty Bonds	$ 7,000.00	
New York City 3½% — 1954 (Par $45,000)	39,696.81	
St. L., Peoria & N. W. 1st 5% — 1948 (Par $10,000)	10,613.89	
United New Jersey Canal 4% — 1923 (Par $10,000)	970.00	
Cash in Banks representing Trust Funds	1,474.16	
		59,754.86

Liquid Assets:			
Liberty Bonds		$48,000.00	
United Engineering Society		2,500.00	
Accounts Receivable:			
Members' Dues	$30,957.06		
Initiation Fees	13,293.25		
Sales of Publications, Advertising, etc.	91,814.75		
		136,065.06	
Cash: In Banks for General Purposes		8,262.14	
			194,827.20
Advance Payments			7,217.85
			$820,532.60

Liabilities

Trust Funds:		
Life Membership	$46,685.39	
Library Development	4,902.71	
Week's Legacy	1,957.00	
Melville Medal	1,330.00	
Hunt Memorial	254.26	
Hess Juniors' Prize	1,000.00	
Hess Students' Prize	1,000.00	
C. T. Main Award	2,625.00	
Total		$59,754.86
Notes Payable		70,000.00
James Watt Memorial		152.60
Initiation Fees Uncollected		13,293.25
Replacement Fund		1,163.18
Steam Table Research		981.21
Dues paid in advance		4,760.64
Unexpended Appropriation 1920–1921		16,700.00
Capital Investment		514,792.79
Surplus and Reserve	$152,583.59	
Less Journal Development deferred account	13,649.57	
		138,934.02
		$820,532.60

The business of the Society has increased so rapidly during the last few years that the savings per year have not furnished quite sufficient cash working capital. A great rate of saving would cut down the general activities to an undesirable degree. Consequently, it must be understood that with more than half of the $150,000 surplus and reserve tied up in stock on hand of publications, etc., and $136,000 accounts receivable, most of which was billed immediately prior to the closing of the fiscal year, the $70,000 notes payable is nothing but an accommodation for thirty to sixty days till the dues of the new fiscal year are received. On the other hand the interest received on deposits of the dues regularly exceed that paid out in interest on notes.

MEETINGS OF THE SOCIETY IN 1921

During the year the standing Committee on Meetings and Program in charge of the Annual and Spring Meetings has adopted the plan of inviting the new Professional Divisions to furnish programs for special technical sessions. To carry out this plan, conferences of representatives of the Professional Divisions have been called several months before each such meeting, and this has developed better professional papers and stronger and more correlated programs.

This Committee, acting with the Standing Committee on Local Sections, carried out the special meeting of the Society to celebrate the Fortieth Anniversary of the Society. The program was devoted to The Opportunities and Responsibility of the Engineer in the Field of Industrial Relations.

The Spring Meeting was held in Chicago, Ill., May 23 to 26. The meeting was well attended, even in the midst of serious business depression, approaching, if not quite at, the bottom of the reaction from the World War. The sessions taxed the facilities of one of America's greatest hotels, the Congress, thereby demonstrating the growth of the Society.

One of the long-to-be-remembered events of the Chicago Meeting in 1904 was the reception given by our beloved Past-President and Honorary Member, Captain and Mrs. Robert W. Hunt. It is, therefore, an event of special note that this event could be repeated in 1921.

Responding literally and loyally to the request of the Council for economy in entertainment in keeping with the times, the Chicago Committee kept expenses down to the lowest in many years and returned to donors considerably over half their contributions.

THE PUBLICATIONS DURING THE YEAR

The Standing Committee on Publication and Papers reported a successful year, despite continued high costs of all publication work.

Transactions. Under instructions from the Council the work on Volume 42 of the TRANSACTIONS for the year 1920 was held back and the volume will be distributed in the Spring of 1922. The reason for this has been entirely the financial situation, and to partially relieve this the Council has authorized a change in future in the distribution of TRANSACTIONS by providing that each member re-

ceive a paper-bound copy free but that a charge be made for any other kind of binding.

Mechanical Engineering. This publication is developing steadily in practically the same form, the leading divisions being technical papers presented at general meetings or contributed, the *Engineering Survey*, and the *Engineering Index Monthly*. During the year a department of signed editorials has been added.

A.S.M.E. News. The Council also accepted the recommendation of the Publication and Papers Committee to convert part two of MECHANICAL ENGINEERING into a separate publication to be called the A.S.M.E. NEWS. This is to be a semi-monthly publication, and the first issue was published on December 22, 1921. The publication will keep the members still more frequently and closely in touch with the work of the Society, including the employment service.

Year Book. Partly on account of the six months' delay in the issue of the previous YEAR BOOK due to a general strike in the printing industry and as another measure for helping the financial situation, the Council delayed the 1921 YEAR BOOK, and an edition was issued in February 1922.

Condensed Catalogues. This has now established itself as an especially successful publication and the Eleventh Annual Volume was issued in the same size, 6 x 9 in., as the previous volumes. However, the Council has approved a plan to enlarge the page size to 9 x 12 in., which will be done for the next volume. This enlarged page size has been under contemplation for many years.

Engineering Index Annual. The third volume published by the Society, much enlarged, was issued uniformly with the preceding volumes. The subjects are arranged alphabetically throughout, and this form of classification seems to meet with general approval. On account of high costs of labor and materials, the price of this valuable publication had to be increased, with regret, but it is hoped in the near future to get back to former prices.

The Council takes occasion to here call attention of the membership to the work of the Publication and Papers Committee, performed with an enterprise which results in but little net expense to the Society.

GROWTH OF THE MEMBERSHIP

This Committee has been extraordinarily faithful and has held thirty-two meetings during the year. From October 1, 1920 to September 30, 1921, 2712 applications were received. The following table details the operations of the Committee for the year:

Applications pending October 1, 1920		1305
Applications received during fiscal year 1920–21		2712
Total applications acted upon 1920–21		4017
Recommended fo Membership		3405
New Applications	3389	
Reinstatements	16	
Promotions Denied		18
Indefinitely Deferred		75
Applications Pending		719
Total Cases Acted Upon 1920–21		4017
Reconsiderations		80
Those recommended for membership were divided into the following grades:		
Members		852
Promotions to Member		266
Associates		99
Promotions to Associate		10
Associate-Members		1001
Promotions to Associate-Member		186
Juniors		991
Total		3405

The membership as of October 1, 1921 was 15,203, an increase of 2468, or of 19 per cent over the preceding year. This is the largest increase, both absolutely and relatively, in any year and the result has been possible only by the loyalty and coöperation of the Membership Committee in continuing the strictest scrutiny of the applications by holding sufficient meetings to pass upon the many applications. The increase in membership may be regarded as a barometer of the virility of the Society, and the figures in the following table will be of interest:

Date	Membership	Date	Membership
1880	189	1901	2254
1881	219	1902	2419
1882	294	1903	2573
1883	364	1904	2619
1884	440	1905	2780
1885	557	1906	2959
1886	602	1907	3006

Date	Membership	Date	Membership
1887	714	1908	3367
1888	857	1909	3455
1889	985	1910	3832
1890	1127	1911	3978
1891	1344	1912	4115
1892	1500	1913	4542
1893	1609	1914	5394
1894	1662	1915	6142
1895	1743	1916	6931
1896	1748	1917	7704
1897	1799	1918	8720
1898	1876	1919	10189
1899	1929	1920	11882
1900	2064	1921	15203

Honorary Membership has been conferred by the Council upon

Nathaniel Greene Herreshoff, for his attainments in the field of naval architecture

Henry R. Towne, Past-President, for his attainments in the field of mechanical engineering

Marshal Ferdinand Foch, for his attainments in the field of military engineering. (The Council is pleased to record that in coöperation with the A.S.C.E., A.I.M.&M.E., and the A.I.E.E. Honorary Membership to Marshal Foch was accorded simultaneously in each Society and presented in joint session.)

In Memoriam. The Society has lost by death one of its Honorary Members, Sir Douglas Fox, whom many members will recall pleasantly in connection with the visit to Europe in 1900.

The other losses, 87 in all, to the Society by death are recorded in the Necrology published in this volume.

There were 116 resignations during the year.

LOCAL SECTIONS CONTINUE UNIFORM PROGRESS

During the past year eight new Local Sections and a new branch of the Connecticut State Section were established. They were, Central Pennsylvania, Inland Empire, Kansas City, Lehigh Valley, Plainfield, Tri-Cities and St. Paul and Minneapolis, the two latter replacing the Minnesota Section, divided for the convenience of the members. This brings the total number of Local Sections up to fifty-one.

International Relations. In coöperation with the other National Engineering Societies, the Council has developed closer ties with

the engineering societies of other countries. This is also bringing about the establishment of societies composed of members of American engineering societies located in foreign countries. Likewise members of our Society in many parts of the world are meeting for the purpose of organizing sections and thus extending the facilities and service of the Society to engineers in all parts of the world.

The officers of the Society and the members of the Committee on Local Sections devote much time to visiting the Local Sections to bring about closer relations with headquarters and to assist in the development of the Sections. In addition, with this end in view the Council itself has held meetings during the year in Syracuse, Boston, and Mobile, Ala.

NEW PROFESSIONAL DIVISIONS ACTIVITY GROWING

Since September 1920 twelve Professional Divisions have been organized: Aeronautic, Cement, Forest Products, Fuels, Gas Power, Machine Shop, Management, Materials Handling, Ordnance, Power, Railroads and Textiles. The total registration in the Divisions in December 1921 was 7882, or one half of the membership.

To show the degree of coöperation and development in this share of the Society's work, at the Spring Meeting eight of the Divisions held technical sessions, and conducted discussions on papers which were presented by the members of the various Divisions; nine of the Professional Divisions held sessions at the Annual Meeting.

On the invitation of the War Department, under the auspices of the Ordnance Division, members of the Society visited the Aberdeen Proving Grounds on October 7, in connection with the Annual Meeting of the Army Ordnance Association. This was also attended by the Society of Automotive Engineers and proved to be a very interesting occasion.

WORK ON CONSTITUTION AND BY-LAWS

In addition to its work on the new Constitution, already recorded, the Committee is engaged in formulating By-Laws and Rules for presentation to the Council by the time the new Constitution goes into effect.

Two amendments to the existing Constitution held over from 1920 were passed for second reading at the Annual Meeting and ordered to letter-ballot of the voting membership on March 1, 1922. These are on the subject of extending the voting privilege to Juniors and on shortening the time to amend the Constitution.

Provision has been made for the annual election of delegates from our Society to American Engineering Council, the governing board of the Federated American Engineering Societies, and By-Laws were adopted to carry out this election in the same way the regular officers are elected.

Nominating Committee. The regular Nominating Committee met in Chicago just before the Spring Meeting and framed its ballots for the officers of the Society and for delegates from the Society to American Engineering Council.

The candidates were duly balloted on by the voting membership and were elected. The names are as follows:

A.S.M.E. OFFICERS

President — Dexter S. Kimball, December 1922

Vice-Presidents

L. E. Strothman, December 1923
Robert Sibley, " "
E. A. Deeds, " " .

Managers

Sherwood F. Jeter, December 1924
H. P. Liversidge, " "
H. P. Porter, " "
Walter S. Finlay, Jr. (to fill the unexpired term, one year, of Dexter S. Kimball.

Treasurer — Wm. H. Wiley, December 1922

A.S.M.E. Delegates to American Engineering Council

To serve for term of two years

Terms expiring January 1924

Francis Blossom
Chas. A. Booth
Gano Dunn
H. H. Esselstyn
W. S. Lee
I. E. Moultrop
John S. Stevens
A. E. Walden
Perley F. Walker

THE ENGINEERING SOCIETIES' LIBRARY

The Standing Committee on Library also represents the Society on the U.E.S. Library Board. The work of recataloguing the Library has proved to be much greater than it was originally estimated. As a consequence the Carnegie Corporation has come to our assistance and appropriated $20,000 to be available over a period of two years. All the books received since 1919 have been properly catalogued. Standard systems of classification have been adopted and modified to suit the needs.

The Library Service Bureau, which supplies, at cost, search and photostatic copies has entailed a slight loss, due to the business depression. However, this loss and a deficit in cataloguing is being repaid out of drastic savings, and in keeping with similar steps taken by business generally.

AN IMPORTANT YEAR IN RESEARCH

Progress reports have been made by the Sub-Committees on Fluid Meters, Lubrication, Bearing Metals, Permanency and Accuracy of Indicating Instruments, and Vibration of Shafting. The Special Committee on Riveted Joints is being organized.

During the year the Council accepted an invitation from authors of steam tables, manufacturers of steam turbines, users of power generating apparatus and others to lend the auspices of the Society through its Research Committee to direct research to the end that constants for the upper limits of steam tables may be authoritatively determined.

The Committee, or perhaps more accurately, its faithful Chairman, has furnished the Publication and Papers Committee monthly for MECHANICAL ENGINEERING a section giving progress in the work in Engineering Research.

STANDARDIZATION

This Committee has during the year worked very closely with the American Engineering Standards Committee, and has considered and made many recommendations to the Council, among them

1 Acceptance of the invitation to appoint a representative on the Sectional Committee which is being organized by the Society of Automotive Engineers and the Bureau of Standards to prepare a Safety Code for Aeronautics. (The Committee approved the acceptance of the invitation and made recommendations to President Carman for this appointment.) The Council approved the acceptance of this invitation and Professor E. P. Warner was appointed as the representative.

2 Approval was also given by the Council to appoint six members to represent this Society on a Sectional Committee which is developing a Safety Code for Mechanical Refrigeration. These men were previously appointed

by the American Society of Refrigerating Engineers but at the suggestion of the American Engineering Standards Committee, requested this change in designation of six of the committee members who had dual membership.

3 During the year the Elevator Code was completed, approved by the Council and published.

A.S.M.E. BOILER CODE AND INTERPRETATIONS

The Committee has continued its regular meetings for the purpose of rendering interpretations and formulating replies to communications from any source respecting boiler construction and the rules pertaining thereto in the A.S.M.E. Boiler Code. Eight meetings were held during the fiscal year, some being held outside of New York.

The Code is being expanded, and during the year the work of the Sub-Committee on Boilers for Locomotives in formulating a special code for railroad locomotive boilers has been completed.

The work of the Sub-Committee on Miniature Boilers in the formulation of a proposed code for addition to the present Boiler Code has progressed.

Progress on the proposed code on Air Tanks and Pressure Vessels has been delayed to such an extent, due to causes beyond the control of the Society, that the Sub-Committee has found it necessary to resume its work without further delay to obtain full coöperation in the important subject of welding, and a revision of its preliminary draft of 1919 was submitted at the Public Hearing at the 1921 Annual Meeting.

The Sub-Committee on Specifications for Steam Piping, Valves and Fittings to be embodied in the Code, held an important meeting at which the scope of its work has been defined. Under instructions from the Boiler Code Committee, the regular Committee of the Society has already undertaken the standardization of dimensions materials, and the initial work on a proposed specification has been completed. An effort is being made to create a recommended code for the installation of steam piping, which may be incorporated in the Boiler Code.

The revision work of the Boiler Code was begun at the March meeting with a view to preparing for the revised edition of 1922. It is the expectation of the Committee to soon make a public report of the proposed revisions and to invite criticism and comments.

POWER TEST CODES REVISIONS

The past year has been one of considerable progress in the development of the Power Test Codes, and the larger proportion of the nineteen Individual Committees have made material progress toward the completion of their respective reports, and the following Codes have been completed:

1 General Instructions for Test Codes
2 Definition and Values
3 Displacement Compressors and Blowers and
4 Reciprocating Steam Engines.

During the year preliminary drafts of the Test Codes on

1 Boilers
2 Gas Producers

were completed by the Individual Committees and distributed. A draft of the Test Code for Internal-Combustion Engines is now being revised by the members of Individual Committee No. 17.

Complying with the recommendation of the Special Committee on Committees for rotative committees, the Main Committee on Power Test Codes is organizing itself as a Standing Committee of the Society, into five groups of five men each. By this arrangement the Committee will be kept alive by the retirement of five men each year, and the appointment of five men to take their places, the retiring men being eligible for reappointment.

PROFESSIONAL ETHICS

Cases of alleged unprofessional conduct have been informally submitted to the Committee during the year. The Society adopted a Code of Ethics in 1914 which contains the fundamental principles underlying good professional conduct. No provisions, however, were ever made for administering this earlier code. It is to be hoped that the Joint Committee on Code of Ethics now working will not only be able to formulate a satisfactory Code for the whole profession, but will also be able to be successful in setting up machinery for its enforcement.

EMPLOYMENT SERVICE

The Employment Bureau of the Founder Societies was taken over on January 1, 1921 as the Employment Service of the Federated

American Engineering Societies, and has been conducted under such auspices but with the supervision of a Temporary Management Committee comprising the Secretaries of the American Institute of Mining and Metallurgical Engineers, the American Institute of Electrical Engineers and The American Society of Mechanical Engineers. The change in the auspices of the Employment Bureau has greatly broadened the scope of the service as it is now prepared to render service to all the twenty-nine Member Societies of the Federation.

While the present unemployment situation is the result of economic industrial conditions and relief cannot be expected immediately, nevertheless, the Society directly and through its representation in the F.A.E.S. Employment Service is doing everything within its power under the circumstances and existing limited appropriations.

AWARDS OF THE YEAR

The Council made the first award of the A.S.M.E. Medal to Hjalmar G. Carlson for his invention and part in the production of 20,000,000 Mark II drawn steel booster casings used principally as a component of 75-mm high explosive shells and bombs.

Life Membership Award for Best Paper Appearing in 1919 Transactions. The Council made this award to Mr. Forest Nagler for his paper on A New Type of Hydraulic-Turbine Runner, appearing in the 1919 Volume of TRANSACTIONS.

Junior Prize. This award was made to Mr. S. Logan Kerr for his paper on the Moody Ejector Turbine.

Student Prize. The Council made these awards for the two best papers submitted by members of Student Branches during the year ending June 20, 1922, to (*a*) Mr. Karl H. White of the University of Kansas for his paper on Forces in Rotary Motors, and (*b*) to Messrs. Richard H. Morris and Albert J. R. Houston, jointly, of the University of California, for their paper on A Report upon an Investigation of the Herschel Type of Improved Weir.

THE SOCIETY AND THE COLLEGES

The Council has authorized branches at the following Colleges during the year: Cooper Union, New York; Rutgers College, New Jersey; University of Texas, Texas; Agricultural and Mechanical College of Texas, Texas; Swarthmore College, Pennsylvania; Oklahoma Agricultural and Mechanical College, Oklahoma; Clemson

Agricultural College, South Carolina; Lowell Textile School, Massachusetts. This brings the total of Student Branches to sixty-two. A complete list of these Branches is published in the Information Section of the YEAR BOOK.

FOREIGN RELATIONS

In the Summer, on invitation from the British and French Engineering societies, a deputation from the American societies was sent to Europe to exchange courtesies and also to award the John Fritz Medal to Sir Robert Hadfield and to Eugene Schneider. Messrs. Ira N. Hollis and Jesse Merrick Smith were the representatives of our Society and Mr. Ambrose Swasey was the Chairman of the entire delegation. Mr. Charles T. Main and Mr. John R. Freeman also were in the delegation, but assigned to the American Society of Civil Engineers, making five A.S.M.E. Past-Presidents in the body.

On October 10, at the Hotel Pennsylvania, New York, a dinner was given in honor of the members who went to Europe with representatives of the European Society governments present, and further assurances of goodwill were exchanged.

A synopsis of the work of the Society was given the membership at the Business Meeting of the Society at the Annual Meeting. On that occasion there was distributed a pamphlet containing the reports of the Standing Committees of the Council in full and this pamphlet has been abstracted in the A.S.M.E. NEWS. The reports of these Committees indicate a successful year of operation with several activities continuing to develop at a very rapid pace. There is a steady growth of interest and coöperation on the part of the membership, and in closing this report the Council expresses its appreciation to all individuals, committees and other bodies who have assisted in carrying the work forward through an eventful year.

No. 1802

PRESIDENTIAL ADDRESS, 1921

By Edwin S. Carman, Cleveland, Ohio
President of the Society

BEFORE discussing the main theme of my address I desire to relate some of the principal Society activities of the past year.

In the year 1920 the spirit of unrest that characterized the human mind as a result of the years of warfare was at its height, and gave expression in almost all walks of life. It was rampant in engineering society circles, and it is with great pleasure tonight that I give expression to my findings covering the operation of the affairs of our Society.

Our Society is democratic in organization, representative in government; its operation is coöperative; its activities extend into every part of the United States, and are within reach of almost every member; its accomplishments are contributions to the welfare of mankind.

As President I have traveled over twenty thousand miles. I have visited and addressed local sections and business organizations, carrying the message of engineering ability and its relation to the great economic problems of the day.

I have failed to find "Politics," or any group of men, or any one man, governing the affairs of our Society.

The San Francisco member has as "much to say" as the New Yorker; the West-Coast members of Council and Committee are as conspicuous as those from any other section; North and South, East and West, are united.

There is not anywhere apparent group or faction, friction or discord, except the friendly differences of opinion. We are all one large family, its members helping each other, all working for the upbuilding of our Society.

One year ago The American Society of Mechanical Engineers

An Address delivered at the Annual Meeting, December 1921, of The American Society of Mechanical Engineers.

celebrated its Fortieth Anniversary. The Local Sections in forty cities held meetings, and with glad hearts told of the accomplishments of the past forty years.

> Forty years of Society activity, and
> Forty cities celebrating
> Forty — a cardinal number with great significance.

The "Fortieth" Anniversary is used many times to indicate completeness. Israel's three greatest kings each reigned forty years; the Ancient Venetian Court was composed of forty.

Do these first forty years represent the First Epoch in the history of our Society?

Has the epoch been completed?

All indications point that way; and if it has, then we are now closing the first year of the Second Epoch. In the years ahead, I believe there will come the realization that a year ago marked the distinct closing of the First and beginning of the Second Epoch.

The important accomplishments of the First Epoch related principally to scientific advancement of material things. What is to be our aim for the Second Epoch?

The beginning of the First Epoch was coincident with the beginning of a constantly and greatly increasing demand for mechanical equipment and allied engineering projects.

The American Society of Mechanical Engineers has been a splendid co-partner in all the technical and scientific developments. It has rendered, and still is rendering a valuable service to industry.

It was, at its beginning, organized to assist in the solution of the great problems of its day. These problems were technical and scientific in their nature, and their solution contributed to the rapid increase in quantity production, through the aid of mechanical appliances. So rapid and complete was the development that many operations were accomplished with little effort on the part of man. But in many instances the elimination of the effort produced an operation of drudgery; and drudgery always creates dissatisfaction and strife.

If we are to follow the policies of the founders of our Society, we too must assist in the solution of the greatest problems of our day.

You will agree with me, I am sure, that the greatest problem of today is neither technical nor scientific, as applied to material things. It is one of human relationships.

If the problem were only the relationship existing as man-to-man, then it would not be one to be considered by engineers. But our problem is more than one of man-to-man, it involves not only the relation of man-to-man, but also the relation of man to production, and man to intense and mechanical production, of man to waste of production and consumption, of man and his relationship to industry, of industry and its relationship to man.

In the early developments, engineers as a whole gave attention, in machine design, or in plans for large projects involving labor, to the producing of results, regardless of their effect on the person or persons engaged as operators or workmen.

Today it is, and it should be, necessary to consider the human side of quantity production, and since this can best be done in connection with technical and scientific progress, it follows that the engineer, whether he so elects or not, must be the one to assume the responsibility, as the technical, scientific, material and human elements of progress must be considered collectively.

Our problem of today had its beginning in the introduction of machines and mechanical appliances to production; with the introduction of the labor-saving devices, there also began that kind of intense, nerve-racking, vitality-consuming labor that has been constantly continuing and increasing, until it has produced one of the greatest problems that has ever vexed mankind. This problem is commonly called "Industrial Relations."

It is true that the origin of labor trouble can be traced back through centuries, but those are not the labor problems that give expression in organized labor today.

Coincident with the demand of civilization for great quantities of the articles of its consumption came the necessity for larger industries. As these industries became more numerous many of them combined and formed large organizations for the purpose of self-protection, self-advantage, and control of markets.

Likewise the bringing together of many workmen, each with a certain and definite task to perform, gave opportunity for an interchange of grievances, and a combined or group effort, — which has resulted in the labor organizations we have today.

Thus we had, growing side by side, two distinct organizations or classes, — one, Industry or Capital, — the other, Labor Unions or Labor. Not only have both of these organizations increased in size, but each has combined and federated with others of its kind, until both are unwieldy, — overgrown, — unable to control or direct

their followers, — each seeking the advantage over the other, and both actually taking advantage of the great consuming class.

We have still another complication, a third party, — the Government, attempting to direct the other two, and the result is more deplorable than before.

Since Government is administered by a class which we in America call "Politicians," and since each class has its own interests to protect, it follows that no one class should attempt to control the other two.

What, then, is the solution? After years of struggle, after countless endeavors from many different angles, the problem remains unsolved, and is growing worse. There must be some method that will solve it.

The party most vitally interested, when rightly lead, will be the deciding factor, — *public interest is paramount.* What group of men is best fitted for leadership?

The great depression of today has seemingly lessened the apparent necessity for urgent consideration of the problem, — but the problem is still with us, unsolved, and with the revival of business it will become more acute than ever.

I recognize in society today the existence of three laws, — the *Moral*, the *Political*, and the *Economic*. I briefly define them as follows:

The Moral Law, God's law, immutable, unchanging
The Political Law, man's law, continually changing
The Economic Law, the law resultant from the operation of the other two.

These laws are co-existing. Their scientific application and operation are necessary; and to have an equilibrium, all of them must function properly and in harmony, just as the various parts of a properly working machine must all function. There is a relationship existing between these laws, just as definite as the proportional relationship existing between the elements that go to make up the strongest bar of alloy steel, the parts of a completed machine, or the departments of a well-organized and successfully operating business.

The Political and Economic Laws must conform to the Moral Law; it is the basic law, and only in this way will complete harmony be secured. These three laws have all been and are in continuous

operation, but they fail to function as a unit, and their independent operation has produced the chaotic condition existing today.

There has been much effort in the last decade to solve the problems of so-called "Industrial Relations." Many plans have been advanced and tried, and while there has been some improvement, nevertheless these plans have fallen far short of their goal. Let me lay special stress on this fact, — any effort, any endeavor, is predestined to failure unless its fundamental principle conforms to each and all of these laws. The world has been asking for thousands of years, "Am I my brother's keeper?" The answer comes back through the ages, "Yes. Love thy neighbor as thyself."

What is the remedy? There must be a speedy adjustment of affairs vital to mankind; we must secure a recognition of and obedience to all of these laws, or we shall soon be in the midst of the greatest conflict of all ages. We are only just beginning to admit that here is the source of most of our difficulties.

If civilization is to continue, and I am sure it is, this complete conformity must be sought for, found, accepted, and reduced to practice. If the search is not to be made by engineers, then it must be made by some other group of men or by some one man. The world awaits the solution; men's eyes are turned, looking with expectancy, some hopefully, some fearfully. They long for peace, the peace that will come only when this age-old struggle between capital and labor has ceased.

Engineers, even though this message has been heralded throughout the land, are slow to accept the challenge to further serve mankind. They are seemingly content to perform only a part of the work for which they have prepared.

The Federated American engineering Societies, speaking for the American engineers, defined "engineering" thus:

> "Engineering is the *science* of *controlling* the forces, and of *utilizing* the materials of nature for the *benefit of man*, and the *art of organizing* and *directing human activities* in connection therewith."

This definition of engineering clearly sets forth the dual nature of engineering; for years we accepted only the single, that of controlling the forces and utilizing the materials of nature, but it is clearly evident that hereafter, neither the material nor the human elements can be considered independently.

In the preamble of the same constitution we find the following:

> "*Service to others* is the expression of the highest motive to which men respond and duty to contribute to the public welfare demands the best efforts men put forth."

If we accept this definition and preamble, and as a Society we have already done so, then we cannot escape our duty. The door is open wide. The opportunity for service is before us, — a service greater by far than the glorious achievements of the past, — an opportunity of weaving together the forces and materials of nature and human efforts, into a composite, yet unified and synchronized unit of production, operating with satisfaction to the worker, to the capitalist, and for the benefit of all mankind.

No. 1803

HEAT BALANCE IN STEAM POWER PLANTS

INTRODUCTION

THE general topic set for discussion at the 1921 Annual Meeting of the Society was the subject of "Waste," which is of vital importance in this country where there is such a widespread disregard of the conservation of its natural resources. The Power Division was asked to confine its session to some feature of this general topic.

Much has been written concerning the design of power-generating equipment, combustion methods, and general design of power stations. Comparatively little, however, has been said concerning the heat balances that have been secured, and that may be secured in steam power plants.

By heat balance we are here to understand that correlation of events in the heat cycle of the plant as a whole which adjusts the rate of steam generation to the demand with the least possible waste of heat, and at the least possible expenditure of coal. The securing of this proper adjustment is the problem now under consideration.

Labor-saving methods and devices have been introduced to such an extent that the cost of fuel is now by far the greatest item of expense in generating power. Improvements in steam turbines, increased efficiencies of boilers, pumps and other equipment have assisted materially in reducing this item. But it still remains the duty of the power-plant designer to design the plant, and to select and install its equipment in such a way that the heat generated therein may be utilized to its fullest extent and every source of waste avoided in so far as possible, in order that his share in the reduction of fuel cost may be performed. Some methods, while apparently very efficient, have inherent losses, which can be determined only by very careful study. There are four principal sources of heat waste with which the power-plant designer has to deal:

1 Radiation
2 Stack losses
3 Losses in condensing water
4 Losses due to excess exhaust steam.

NOTE. — The Society as a body is not responsible for the statements of facts or opinions advanced in papers or discussions (C 55)

The means which he employs to reduce these losses determine the heat balance to be secured in the operation of the plant.

With the thought that much good could result from a general discussion of the subject of heat balance, engineers from various companies were asked to present the ideas used in the design of some of the recent stations.

Intimately involved in the solution of the heat balance is the method employed for driving the station auxiliaries. In the early plants they were driven principally by means of steam engines of high water rates. Improvements in steam-engine design and the introduction of high-speed turbines gradually increased the economy of the steam-driven auxiliary systems; later the electric motor, both alternating and direct-current, found favor with designers. Today there are advocates of many systems of auxiliary drive. Some engineers prefer to use only steam drive, others alternating current, some direct current, and many favor combinations of steam and motor drive.

The Connors Creek Station of the Detroit Edison Company has been in operation for several years, and figures are given in one of the following papers concerning the results obtained in this station, in which the auxiliaries are motor-driven, being supplied with power from house alternator units.

The station of the Philadelphia Electric Company is an example of combination motor and steam drive, with electric power supplied from the main station bus. In this station economizers are used, this being a departure from the practice of other stations of which accounts are given.

The Colfax Station of the Cheswick Power Company has a combination auxiliary system of steam and motor units supplied from house alternators.

A brief description has been given of the system to be used at the new Hell Gate Station of The United Electric Light and Power Company in New York. This station will use almost exclusively electric drive, with one-half the power obtained from the main generators and half from the house alternators, with no provision for interconnecting the two sources of power.

The systems used in the four stations mentioned above are the embodiment of the ideas of their respective designers as to the best methods of properly working out the heat balance for these stations.

No. 1803 *a*

AUXILIARY SYSTEM AND HEAT BALANCE AT THE DELAWARE STATION OF THE PHILADELPHIA ELECTRIC COMPANY[1]

By E. L. Hopping, Philadelphia, Pa.
Member of the Society

FOR the past few years the high cost of fuel and other materials used for the generation of electrical energy, together with the high cost of labor for operation, has made it imperative that the maximum economy should be obtained in the use of materials and labor entering into the construction and operation of the public-utility company's steam-electric generating stations. The greatest economies are to be found in the efficient burning of the fuel for steam generation and in the proper utilization of the energy contained in the steam. There are various factors entering into the problem in connection with the auxiliary system and heat balance for any of the large generating stations that must be considered. Any decision regarding the auxiliary system and heat balance involves the consideration of certain outstanding conditions, which may be classified in the order of their importance as follows:

a Reliability of service
b The economy in heat units per kilowatt of net output
c Simplicity and flexibility of operation with which the heat balance may be maintained under widely varying conditions of load and temperature
d The investment and maintenance cost of the equipment used in obtaining the desired economies, and its comparative value to other systems in terms of income from the investment
e The flexibility of the system in view of possible future developments and changes in power-plant equipment.

2 In working out the auxiliary system and heat balance for the Delaware Station of The Philadelphia Electric Company, the

[1] For discussion and closure see p. 516.

Presented at the Annual Meeting, December 1921, of The American Society of Mechanical Engineers.

factor of reliability of service was considered of prime importance, and of somewhat greater value than that of heat economy. A study of the system will indicate its flexibility and its value in meeting varied conditions or varied temperatures in boiler feedwater which might be obtained with the installation of apparatus such as air extractors.

3 It has always been considered by this company that the small comparative cost of duplicating the main-generating-unit auxiliaries is fully warranted in view of the resulting ability to keep a large unit on the line in case of failure of any one of the different auxiliaries connected with the unit. This particular item is one which must be considered almost entirely in the light of reliability of service, since no direct returns on the investment can be shown by the installation of duplicate auxiliary equipment.

4 Having the question of duplication of auxiliaries decided, the next step was to settle upon the method of driving the various auxiliaries in the plant. Due to temperature changes in the circulating water, the vacuum on the main generating units varies from about 28 in. during the summer months to 29.25 in. during the winter months. Temperature charts of this water condition show average variations from 40 to 80 deg. fahr. With this variation in vacuum there must be considerable flexibility in the method of obtaining the proper amount of heat for bringing the temperature of the feedwater to the required point, which in this case, up to the present time, has been 210 deg. fahr. Knowing the amount of heat required under varying vacuum and load conditions, it remained to select such auxiliaries to be steam-driven as would furnish the required heat through their exhaust. It was found possible to obtain this result by installing turbine drive on one of each of the duplicate auxiliary pumps connected with the main generating units, together with turbine drive on boiler-feed pumps and on one house pump. It was found necessary, however, at times of heavy load and high vacuum, to supply a certain additional amount of heat, which is obtained by bleeding the main turbine at the eleventh stage.

5 By reference to Fig. 1, which shows the arrangement and the type of drive of the auxiliaries for both the boilers and turbines, it will be noted that all of the boiler-feed pumps are turbine-driven. This practice has been followed throughout the system of The Philadelphia Electric Company and has been made standard, based upon the fact that in case of a serious disturbance in the electrical system of a particular plant, the boiler-feed pumps would not be

affected. At such a time it is very necessary that these pumps be available for duty.

6 The forced- and induced-draft fans and the stokers are motor-driven and receive their current through a station light and power system. These auxiliaries and their protective equipment, described more fully later on in this paper, would be shut down only in case of an interruption of the service to the two buses of this system.

7 On each of the main units one of the two circulating pumps is installed with a duplex (steam and motor) drive By use of this type of drive two results are obtained:

a A flexible unit is available, which, by varying the load on the turbine drive, will supply the required heat variation for maintaining a practically constant feedwater temperature

b In case of trouble in either the steam supply or the electric service, the opposite drive will automatically pick up the load, and thus keep the condenser water in circulation. The other circulating pump, which is provided with a motor drive only, is used as a reserve, or in case it is desirable to operate two circulating pumps at a time when high circulating-water temperatures occur.

8 For the air and condensate pumps a duplex drive is not deemed necessary, as, in case of trouble with either of these auxiliaries a brief shutdown would not be serious. If one of the pumps is in trouble, it only requires a very short time to put the other pump in service. During this time the vacuum, in case of an air-pump failure, should not be affected to any appreciable extent, and the condensate storage capacity in the condenser would take care of the time element in getting the other condensate pump on the line.

9 Throughout the plant apparatus such as coal hoists, conveyors, miscellaneous pumps, etc., are motor-operated. This equipment is all for intermittent service, and a brief shutdown of any piece of apparatus would not affect the plant system. In only two cases have we installed turbine drive on miscellaneous pumps. These are in the case of the fire pump and the house water-service pump. Turbine drives were installed on duplicate pumps in these instances in order to protect the plant should an interruption to the electrical service occur.

ELECTRICAL SYSTEM

10 While considering the extensive use of electric drive for auxiliaries, it was clearly realized that it would be unwise to take advantage of the economies thus made possible unless the reliability of this supply and the sturdiness of the equipment were comparable to steam drive. In order to insure this, four considerations were kept in mind in the design of the station light and power system, namely:

a Great care must be used in the selection and layout of the equipment, so as to give a high degree of reliability

b In case of trouble on the system any defective piece of apparatus must be isolated immediately with the minimum effect on other apparatus

c Any apparatus whose operation might have been interfered with must be restored to service as quickly as possible with the minimum of attention by the operating force

d Any disturbance on the main generating system must have a minimum effect on the station light and power system.

11 The electric load was found to be of such proportions that had low voltage (600 volts or less) been employed the investment and losses would have been very great. This consideration led to the adoption of 2300 volts for the supply of the larger motors. Though there was no definite experience available on which to base such a course in central-station practice, the economies thus effected, together with the improvements which had been made in 2300-volt equipment, resulted in the use of this potential for motors of 100 hp. and larger. Since ruggedness and reliability are essential for service on boiler and turbo-generator auxiliaries, it was recognized that great care must be exercised in selection of the 2300-volt equipment. With these thoughts in mind, features such as especially rugged contactors, both primary and secondary, for slip-ring motors, increased spacing of slip rings, 40 deg. cent. motor ratings, and oversize starting compensators were included in the specifications.

12 In the design and construction of the 2300-volt station auxiliary system the same care was exercised as in the case of the main 13,200-volt system. Busbars and oil circuit breakers are housed

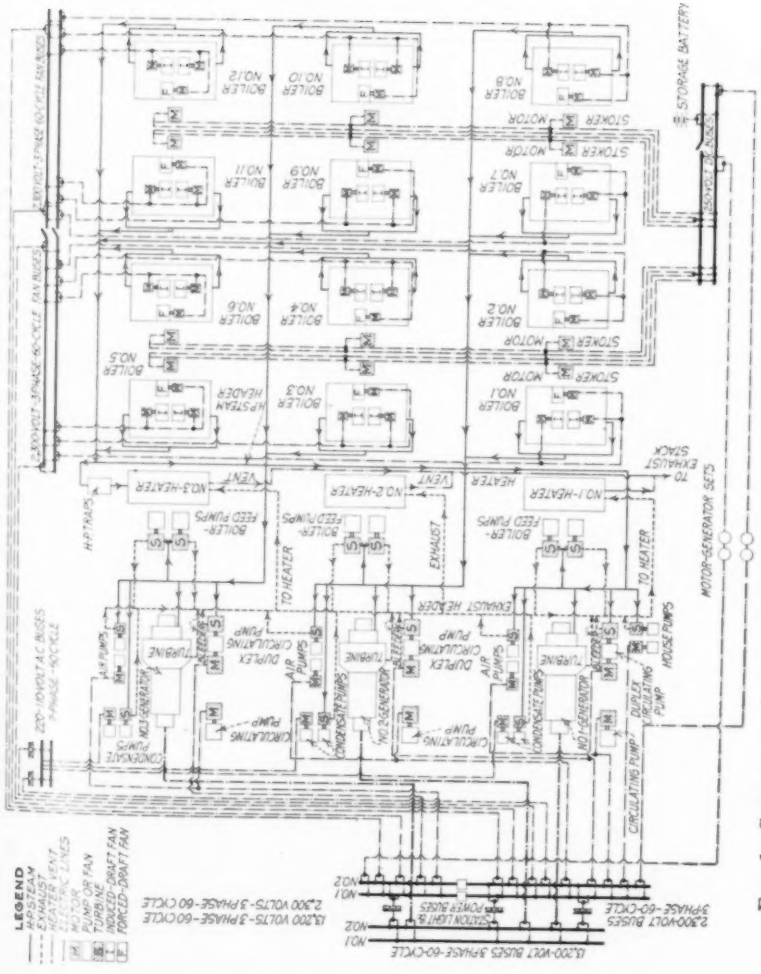


FIG 1 STEAM AND ELECTRIC DISTRIBUTION DIAGRAM FOR THREE 30,000-KW. UNITS

in concrete compartments, and breakers of high rupturing capacity are used. By the use of differential relays on all of the important motors, assurance is given that the latter will not be disconnected

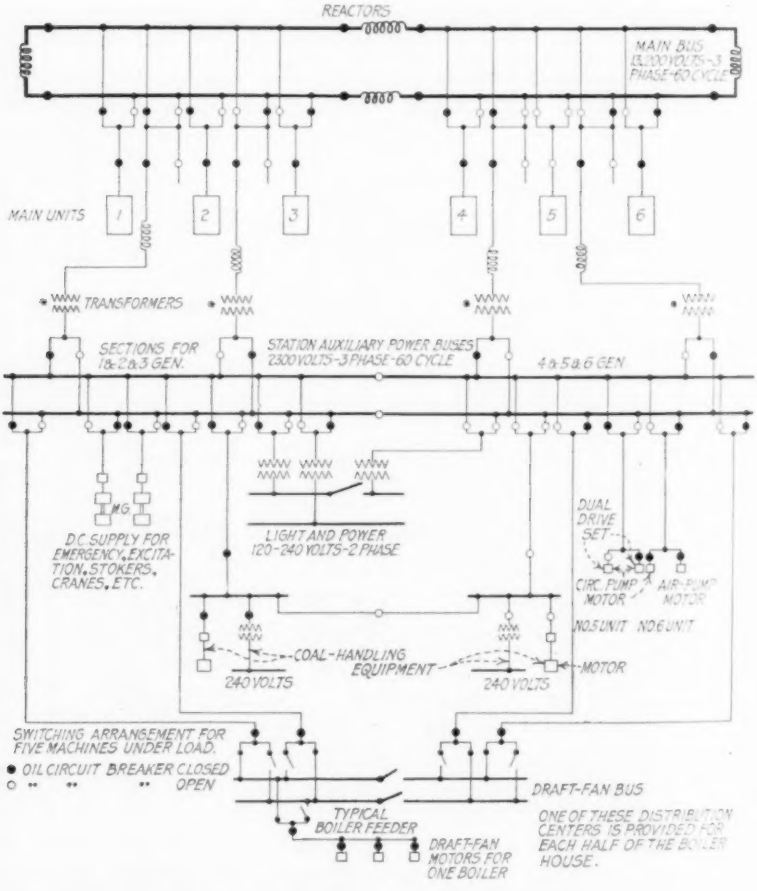


FIG. 2 STATION LIGHT AND POWER SYSTEM

(Entire auxiliary power load can be carried by three of the four transformer banks. Disconnecting switches are shown only where they are used for sectionalizing or selecting buses. The only auxiliary power feeders shown are those supplying centers of distribution. Differential relay protection on transformers (marked *), condenser, auxiliaries, draft-fan motors and their cables.)

from the line for any reason other than failure within themselves or of the cable supplying them, thus insuring continued operation

during system disturbances. The motors used with the boiler fans and circulating pumps are so arranged that, in the event of an interruption to service, they will automatically be restored to the condition of operation existing previous to the interruption.

13 It was recognized that boiler-room attendants and even turbine engineers would probably not be familiar with electrical equipment, and for this reason and for greater reliability of operation all apparatus is of the remote-control type. Hand-reset relays in locked compartments are employed, so that it will be impossible for any one but the plant electrician to restore service to a motor that has failed, and more than ordinary precautions are taken in enclosing all electrical equipment that is located in the boiler and turbine rooms.

14 Fig. 1 illustrates the steam and electrical distribution for three 30,000-kw. units, although Nos. 1 and 2 units are the only ones at present installed. The capacity of this plant has been planned for an ultimate installation of six 30,000-kw. generating units, and the electrical distribution as described below will take care of the complete installation.

15 Reference to Fig. 2 will indicate the switching arrangement for the station light and power system. The electrical supply to auxiliaries will be through four 5,000-kva. 13,200 to 2300-volt transformer banks. Three of these are ample to carry the ultimate peak load. This energy is distributed to the various auxiliaries through twenty-eight 2300-volt feeders as follows:

9 feeders to circulating- and air-pump motors
8 feeders to boiler-fan motors' distributing bus
2 feeders to coal tower
2 feeders to motor-generator sets for d.c. supply
1 feeder to motor-driven exciter
2 feeders to miscellaneous motors' distributing bus
3 feeders to low-tension transformers
1 feeder to test set (for testing high-tension cables).

There will be three 1000-kva. transformer banks for stepping the voltage from 2300 volts three-phase down to 115–230 volts two-phase to supply small motors and lighting. There will also be two 300-kw. motor-generator sets to convert a.c. energy to 125–250 volts d.c. for driving motors of wide speed variation, such as those used for stoker drive. A storage battery serves as an emergency supply for this equipment.

MOST ECONOMICAL HEAT DISTRIBUTION

16 Two heat-distribution diagrams, Figs. 3 and 4, have been worked out, based on the present installation of two 30,000-kw. generating units. Fig. 3 illustrates the heat distribution for an hourly load at the most economical point of operation for the turbine, and with a vacuum of 29 in. Nearly all of the values shown are based on the results of actual tests conducted on the apparatus since its installation, and the coal rate has been based on a heating value of 13,500 B.t.u. per pound of coal as fired. At the time the heat balance for this plant was originally calculated, coal received contained an average heating value of 13,500 B.t.u., as used in this figure. This value has been used in calculations for all preliminary, or theoretical, heat balances.

17 The overall efficiency of the boiler, stoker and economizer of 80.4 per cent, given on Fig. 3, is based on an actual test made during the month of June 1921, when an efficiency of 81.7 per cent at a boiler rating of 176 per cent was obtained. A correction has been made on this diagram for the lower air temperatures that would prevail during the winter months at which time a vacuum of 29 in. is obtainable.

18 Under "heater vent losses" we have included a loss of about 5 per cent of the exhaust steam used for feedwater heating. This loss would be difficult to measure with any accuracy, but we believe, from rough checks on heat absorbed in feedwater, that the figure of 5 per cent should cover the losses at this point. The economizer and blowdown losses are correct within the limit of accuracy of the V-notch meter used to record these losses.

19 The item of line radiation losses, while apparently small, is noted separately, due to the fact that this is a heat loss only. When the original heat balance was figured for this plant, it was felt that it would never be necessary to bleed the turbine, due to the fact that a large increase in temperature of the condensate was expected as a result of passing the condensate through the waterbacks in the boiler bridge walls. It was found, however, in actual practice that, due to the very large combustion chamber, and the more complete combustion secured therefrom, the temperature rise of the water was almost one-third less than that originally estimated.

20 The heat distribution for the hourly condition (Fig. 3) does not, of course, make any allowances for banking or emptying of boilers for repairs. These are about the only losses which would occur over continued operation which is not included in this diagram.

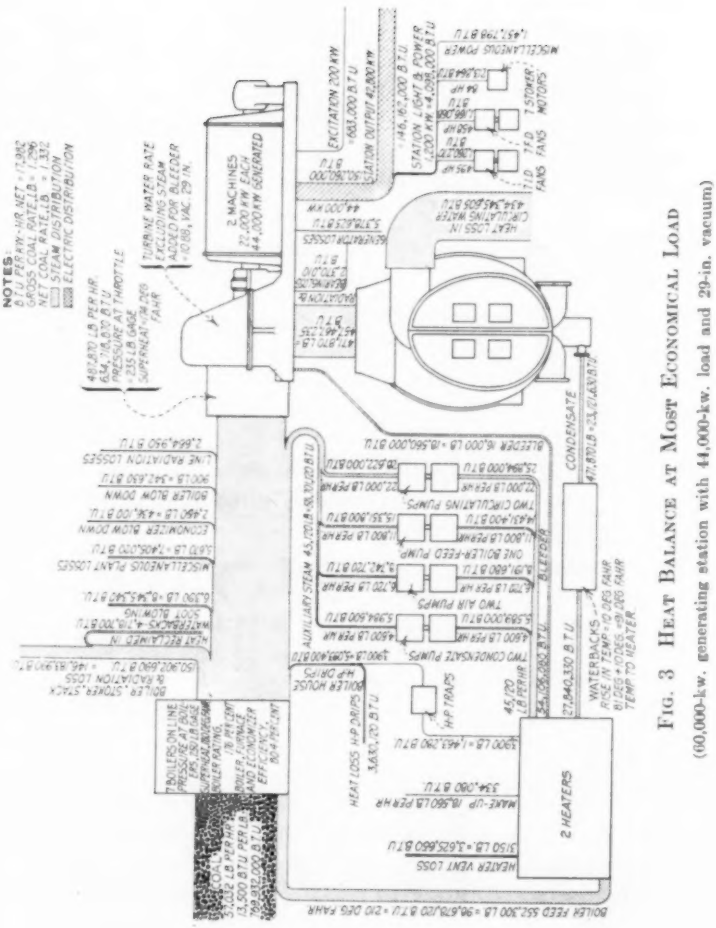


FIG. 3 HEAT BALANCE AT MOST ECONOMICAL LOAD

(60,000-kw. generating station with 44,000-kw. load and 29-in. vacuum)

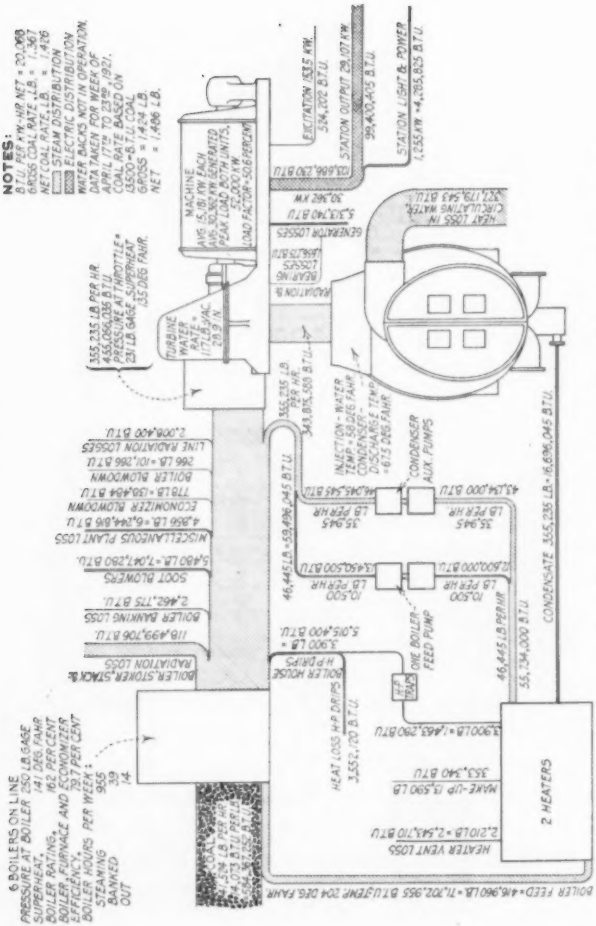


FIG. 4 HEAT BALANCE FOR ONE WEEK'S OPERATION

(60,000-kw. generating station having average load of 30,362 kw. and average vacuum of 28.9 in.)

TYPICAL WEEKLY HEAT DISTRIBUTION

21 Fig. 4 illustrates the actual heat distribution over a period of one week, during April 1921. It will be noted on this diagram that the load factor of the plant, based on its maximum capacity, was only slightly over 50 per cent which means, in this case, that the average boiler efficiency is lower, and the average turbine water rate higher, than will be obtained under more favorable load conditions. Several records of weekly coal rates which very nearly approach this result have been obtained with a lower load factor, and the plant operating over considerable periods of the day with light loads on the turbine.

22 There are several items on this diagram which differ from the diagram in Fig. 3, as follows:

a A considerable difference will be noted between the heat value of the coal as used in this diagram and that used in Fig. 3. The coal was comparatively high in heat value during this week's operation, whereas the average over a longer period would be lower

b All of the turbine-driven condenser auxiliaries have been grouped, as the number of auxiliaries used in varying conditions necessarily varies with the load and temperature

c The station light and power load has also been given as one item, due to the fact that the number of auxiliaries in service at various times depends on the load and temperature

d There is no temperature increase shown in the condensate between the condenser and the heaters, as the waterbacks had not been connected to operate on condensate at that time. The heat which ordinarily would be picked up in the condensate water in this case was lost and is accounted for in the stoker and radiation losses.

e The average boiler-feed temperature at this time (for the diagram in Fig. 4) was 204 deg. fahr. It was found in operation that this temperature is too low, and that not enough oxygen is liberated in the heaters to allow for operation of the boilers without considerable pitting in the integral economizer tubes

f The amount of superheat in the steam at this time was lower than shown on the diagram in Fig. 3, due largely

to the fact that, with the combustion conditions obtained in the large furnaces, the superheat was below guarantees. Since that time, the superheat has been increased by changes in baffling, with resulting increase in efficiency on the turbine water rate.

23 Throughout the week illustrated in this diagram there were six boilers on the line, with the exception that over the week end, when the station load is usually low enough to be carried on one turbine, one boiler was taken out for repairs and cleaning. The banking hours represent the hours during which certain boilers were banked to take care of low-load conditions between midnight and 6.00 a.m. It was found more economical to bank boilers for this period than to operate the entire six boilers at a lower rating.

24 Summarizing the heat-balance situation at this station, it is believed that, while there may be a slight loss in heat economy as compared with some other methods of obtaining heat balance, the very important item of reliability of service as obtained by this system, and the ease with which it may be operated will offset the possible gains in heat economy secured by other systems.

25 At the present time steps are being taken toward the installation of an air extractor, which, by removal of the oxygen from the feedwater, will not only protect the piping, economizers and boilers from corrosion due to this cause, but will increase the boiler efficiency and the overall plant efficiency by allowing operation with feedwater temperatures of about 140 deg. fahr.

26 In the operation of this auxiliary system the only points of variable control which must be taken care of by the operator are the steam supply through the duplex circulating pump and the bleeder steam. These are both, at the present time, manually operated by the regular pump-room engineers. All of the other steam-driven auxiliaries are operated to suit the larger changes in plant load and temperature conditions. These changes do not come very frequently, and are fairly regular in the time at which they occur, so that this phase of operation of the auxiliary system is of only minor importance.

27 The author wishes to extend his thanks to R. A. Hentz, assistant electrical engineer of The Philadelphia Electric Company, for assistance in the preparation of the electrical features of this paper, and to the other associates in the engineering department who assisted him in assembling the data.

No. 1803 *b*

HEAT BALANCE OF COLFAX STATION[1]

By C. W. E. Clarke, New York, N. Y.
Member of the Society

IT has been the practice for a good many years to drive certain auxiliary station apparatus by steam and to use the exhaust from this apparatus for heating the feedwater. Within recent years various attempts have been made to proportion the amount of steam-driven auxiliary apparatus so as to have only enough exhaust steam to maintain the feedwater at a suitable temperature. This system has taken on many different forms, but the purpose of all of them is to avoid having any excess of exhaust steam that would have to be wasted. Systems having the foregoing object in view have received the general name of "Heat-Balance System." It is not within the scope of this paper to discuss the relative merits of various types of heat-balance systems. The author will simply endeavor to explain briefly, but as clearly as possible, the system adopted for the Colfax Station of the Duquesne Light System, located at Cheswick, Pa., and give some of the results so far attained with it.

2 The Colfax Station is designed for an ultimate gross capacity of 360,000 kw. and for a maximum output of 300,000 kw. The generating equipment as now planned will consist of six 60,000-kw. turbo-alternators, of which one is now in service and a second in process of installation. The immediate boiler equipment will consist of seven B. and W. cross-drum boilers, each having 20,876 sq. ft. of heating surface, and four others of the same type, each with 22,910 sq. ft. of heating surface, are to be installed in the near future. These boilers will be equipped with 17-retort extra long Westinghouse underfeed stokers. Each of the present units has four 25,000-sq. ft. Westinghouse surface condensers, two for each low-pressure element. All condenser auxiliaries are motor-driven. There are three 1500-g.p.m. turbine-driven centrifugal boiler-feed pumps. The forced-draft equipment consists of three 250,000-c.f.m. Greene fans, turbine-driven through reduction gears. All other station auxiliary apparatus is motor-driven. A 2000-kw. house turbine is provided to supply auxiliary power. This turbine comprises part of the heat-balance system as described below.

[1] For discussion and closure see p. 516.

3 Fig. 1 shows the bus arrangement for supplying the electrical auxiliaries. There are two buses from which the auxiliaries may be supplied, one fed through a transformer bank from the main-station bus, and the other fed by the house turbine. These two buses are tied together through a motor-generator set, so arranged that energy can pass from the auxiliary bus to the house-turbine bus but not in the opposite direction. A direct tie is provided between these two buses for use in case of failure in motor-generator connection. The switches in the direct tie and the motor-generator connection are interlocked so that only one can be closed at a time.

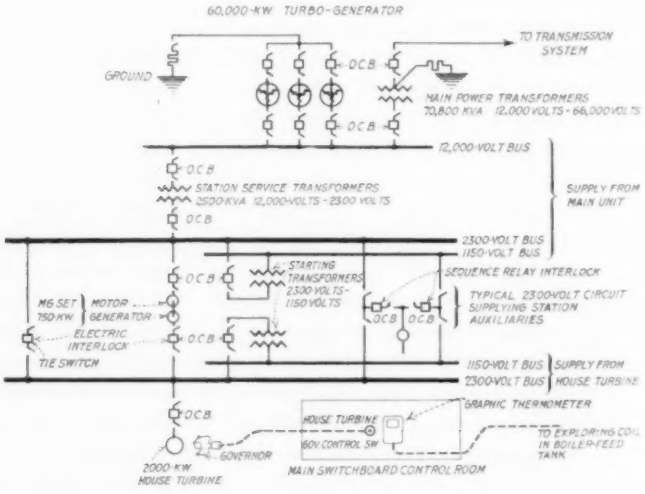


FIG. 1 CIRCUIT DIAGRAM FOR AUXILIARIES AND HEAT-BALANCE CONTROL

Auxiliaries are so connected that they may be fed from either bus, the switches being provided with interlocks so that both cannot be closed at the same time. With the present auxiliary load the auxiliary bus fed from the main station bus is not necessary. The house turbine is ample to carry the entire load. When the second 60,000-kw. unit is installed, however, the auxiliary load will exceed the capacity of one house turbine, resulting in part of this load being placed directly on the auxiliary bus and being fed from the main bus. Only as much load will be thrown on the house turbine bus as is necessary for heat balance.

4 For starting purposes an 1100-volt bus is provided, which obviates the necessity for compensators and other starting devices. The starting and running switches are provided with sequence interlocks, which makes it impossible to close the switch on the 2300-volt bus until after the 1100-volt switch has been closed and opened.

5 Due to the slip in the motor-generator set, Fig. 3, the frequency produced by the generator is about 57 cycles when the main-bus frequency is 60 cycles. If the speed of the house turbine is varied so as to make its frequency slightly above or below that of the house-turbine bus, the amount of load carried by the house turbine will be correspondingly varied. This system of using a

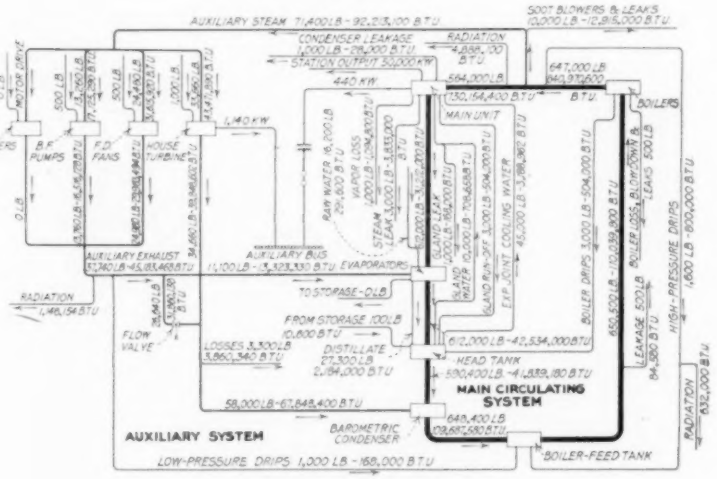


FIG. 2 HEAT-FLOW DIAGRAM

Average vacuum in main condenser, 28.73 in.

house turbine paralleled on the main bus is not new, but in existing installations of this kind considerable difficulty has been met with, due to differing characteristics of the small house turbine and the large units supplying the main bus. The motor-generator set mentioned above was introduced into the system to avoid this difficulty. This piece of apparatus acts somewhat like a flexible coupling, taking up the momentary variations in frequency on both sides. This arrangement permits of any desired variation in the quantity of exhaust steam available for heating the feedwater since the greater part of the steam for feed heating comes from the house turbine.

Its control can be practically accomplished by manipulation of the house-turbine governor control on the main switchboard alone. Manipulation of the barometric injection valve is occasionally necessary. A graphic recorder in the control room indicates the boiler-feed temperature so that the switchboard operator can maintain the boiler-feed temperature at any desired point.

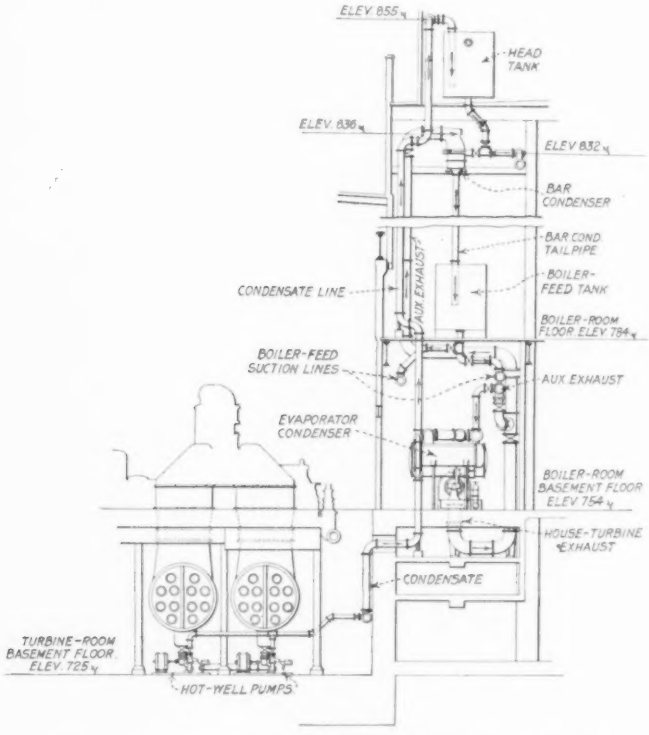


FIG. 3 GENERAL ARRANGEMENT OF FEEDWATER SYSTEM

6 The general heat cycle is indicated on the diagram, Fig. 2. As will be seen from this diagram, the condensate passes from the hot well to the double-effect evaporator, where it acts as a circulating medium in the evaporator condenser, absorbing the heat of vaporization from the water evaporated. Thence it goes to a head tank located on the boiler-house roof. This tank merely acts as a small storage reservoir to take care of variations in water flow in this part of the system. Normally, the make-up to replace water losses

TABLE 1 HEAT BALANCE

Day	A				B			
	Pounds per hour	Pressure, lb. per sq.in.	Superheat, deg.	B.T.u. above 32 deg.fahr.	Pounds per hour	Pressure, lb. per sq.in.	Superheat, deg.	B.T.u. above 32 deg.fahr.
Steam Produced								
Steam to main unit	771,000	287	165	1,002,531,300	648,000	287	162	841,622,400
Steam to house turbine	667,000	260	157	875,631,800	562,000	271	155	727,171,800
• Steam to house turbine	34,300	255	180	44,222,990	34,180	266	180	44,119,962
• Steam to feed pumps	21,400	255	180	27,591,020	13,460	266	150	17,378,514
• Steam to draft fans	27,800	255	180	35,842,540	24,860	266	150	32,091,774
• Steam to soot blowers	10,000	255	180	12,893,000	13,000	266	150	16,781,700
• Steam to drips returned, 532 deg.temp.	500			250,000	500			250,000
Heat loss unaccounted for				6,099,250				3,831,650
Total	771,000			1,002,531,300	648,000			841,622,400
Total heat in steam from boilers				1,002,531,300				841,622,400
Total heat in feedwater				126,901,165				106,552,640
Net heat added in boiler				875,630,135				735,069,760
B.T.u. per Kw-hr. (station load)				14,200				14,600
Main Unit Input								
Steam to main unit	677,000	260	157	875,631,800	562,000	271	155	727,171,800
• Gland water	10,000		97.41	654,190	10,000		96.29	642,920
• Condenser leakage	1,000		55	23,000	1,000		52	20,000
• Expansion joint water	40,000		97.41	2,616,400	40,000		96.29	2,571,680
Total	728,000			878,925,390	613,000			730,406,400
Main Unit Output								
Useful power (main unit load)				221,217,750				172,650,225
• Steam leak hp. gland	3,000	260	157	3,880,200	3,000	271	155	3,881,700
• Vapor loss through air pump	1,000		60	1,094,800	1,000		80	1,094,800
Condensate metered	720,000		74.2	30,384,000	605,000		75	26,015,000
• Gland water leaks	1,000		200	168,000	1,000		200	168,000
• Gland run-off	3,000		200	504,000	3,000		200	504,000
Losses in main unit condenser, etc.				621,676,640				525,892,575
Total	728,000			878,925,390	613,000			730,406,400
Available Exhaust for Feed Heating								
• From house turbine	34,300	14.24	0	39,439,240	34,180	14.24	0	39,296,746
• From house turbine glands	1,000	14.24	0	1,149,700	1,000	14.24	0	1,149,720
• From feed pumps	21,400	18	103	25,737,780	13,460	17	101.6	16,164,114
• From feed-pump glands	1,000	18	103	1,202,700	500	17	101.6	600,450
• From draft fans	27,800	18	103	34,438,060	24,860	17	101.6	29,854,374
• From draft-fan glands	500	18	103	601,550	500	17	101.6	600,450
Total	86,000			101,569,830	74,500			87,665,834
Exhaust Steam Used								
In evaporator	15,000	18	103	18,040,800	11,700	17	101.6	14,050,530
In barometric	69,000	14.24	427	80,716,200	59,000	14.24	42.7	69,018,200
Returned as drips	1,000		200	168,000	1,000		200	168,000
• Loss through relief valve	1,000	14.24	42.7	1,169,800	2,800	14.24	42.7	3,275,440
Losses unaccounted for				1,471,330				1,153,664
Total	86,000			101,565,830	74,500			87,665,834
Evaporator								
Raw water	16,000		49	272,000	17,000		49	289,000
Steam to evaporator	15,000	18	103	18,040,800	11,700	17	101.6	14,050,530
Condensate	720,000		74.2	30,384,000	605,000		75	26,015,000
Total	751,000			48,696,800	633,700			40,354,530
From Evaporator								
Condensate	720,000		96	46,080,000	605,000		94	37,810,000
Distillate to head tank	28,500		115	2,365,500	28,700		110	2,238,600
Total to head tank	748,500			48,445,500	633,700			39,748,600
Distillate to storage tank	2,500		115	207,500	0			0
Total from evaporator	751,000			48,653,000	633,700			39,748,600
To Head Tank								
Condensate	720,000		96	46,080,000	605,000		94	37,810,000
From storage tank	0			0	8,500		140	918,000
Distillate mixture	28,500		115	2,365,500	28,700		110	2,238,600
• Main unit gland run-off	3,000		200	504,000	3,000		200	504,000
• Drips from boiler room	2,000		200	336,000	2,000		200	504,000
Total to head tank	753,500		97.41	49,285,500	648,200		96.29	41,674,600
From Head Tank								
To barometric heater	703,500		97.41	46,014,990	598,200		96.29	38,460,000
• To main unit glands	10,000		97.41	654,190	10,000		96.29	642,920
• To expansion joints	40,000		97.41	2,616,320	40,000		96.29	2,571,680
Total from head tank	753,500			49,285,500	648,200			41,674,600
To Barometric Heater								
Water from head tank	703,500		97.41	46,014,990	598,200		96.29	38,460,000
Steam	69,000	14.24	42.7	80,716,200	59,000	14.24	42.7	69,018,200
Total to heater	772,500		196.08	126,731,190	657,200		195.54	107,478,200
To Feed Pumps								
• Live steam drips returned	500		200	84,000	500		200	84,000
• Low pressure drips returned	1,000		200	168,000	1,000		200	168,000
From barometric heater	772,500		196.08	126,731,190	657,200		195.54	107,478,200
Total	774,000		196.06	126,983,190	658,700		195.55	107,730,200
• Station water losses	500		196.08	82,028	500		195.55	81,775
Total to boilers	773,500		196.06	126,901,165	658,200		195.55	107,648,425
• Boiler water lost	500				500			
• Boiler water returned	2,000				3,000			

TABLE 1 HEAT BALANCE (*Continued*)

C				D				E			
Pounds per hour	Pressure, lb. per sq. in.	Superheat, deg.	B.T.u. above 32 deg. fahr.	Pounds per hour	Pressure, lb. per sq. in.	Superheat, deg.	B.T.u. above 32 deg. fahr.	Pounds per hour	Pressure, lb. per sq. in.	Superheat, deg.	B.T.u. above 32 deg. fahr.
647,000	287	164	840,970,600	561,000	287	153	725,990,100	527,000	287	138	683,453,660
564,000	271	157	730,154,400	492,000	273	150	635,270,400	488,000	275	152	591,944,200
33,660	265	152	43,471,890	20,000	268	144	25,752,000	22,000	270	140	28,342,600
13,260	266	152	17,125,290	13,000	268	144	16,738,800	12,000	270	145	15,459,600
24,480	266	152	31,615,920	21,000	268	144	27,039,600	20,000	270	145	25,766,000
10,000	266	152	12,915,000	12,000	268	144	15,451,200	12,000	270	145	15,459,600
1,600			800,000	3,000			1,800,000	3,000			1,500,000
			4,888,100				4,238,100				4,324,660
647,000			840,970,600	561,000			725,990,100	527,000			683,453,660
			840,970,600				725,990,100				687,465,660
			110,039,800				95,267,700				82,765,379
			730,930,800				630,722,400				599,694,281
			14,600				15,600				15,450
564,000	271	157	730,154,400	492,000	273	150	635,270,400	456,000	275	152	591,944,200
10,000		102.87	708,554	10,000		114.31	822,100	10,000		109.02	770,200
1,000		60	28,000	1,000		72	40,000	1,000		56	28,000
45,000		102.87	3,188,262	0			0	40,000		109.02	3,026,273
620,000			734,080,016	503,000			636,133,500	509,000			595,818,073
			172,252,600				142,268,900				134,892,500
3,000	271	157	3,833,000	3,000	273	150	3,873,600	3,000	275	152	3,877,200
1,000		80	1,094,800	1,000		92	1,100,100	1,000		80	1,094,800
612,000		83	31,212,000	495,000		98.8	31,581,000	501,000		79	22,547,000
1,000		200	168,000	1,000		200	168,000	1,000		200	168,000
3,000		200	504,000	3,000		200	504,000	3,000		200	504,000
			485,012,648				456,437,900				434,734,373
620,000			734,080,016	503,000			636,133,500	509,000			595,818,073
33,660	14.24	0	38,698,902	20,000	14.24	0	22,994,000	22,000	14.24	0	25,293,400
1,000	14.24	0	1,149,700	1,000	14.24	0	1,149,700	1,000	14.24	0	1,149,700
13,260	17.5	99	15,915,978	13,000	17.5	109	15,665,000	12,000	17	127	14,751,600
500	17.5	99	600,150	500	17.5	109	602,500	1,000	17	127	1,211,800
24,480	17.5	99	29,383,344	21,000	17.5	109	25,305,000	20,000	17	127	24,256,000
500	17.5	99	600,150	500	17.5	109	602,500	700	17	127	848,860
73,400			86,348,224	56,000			66,318,700	56,700			67,314,460
11,100	17.5	99	13,323,330	8,700	17.5	109	10,483,500	14,600	17	127	17,706,880
58,000	14.24	42.7	67,848,400	44,600	14.24	42.7	52,173,080	39,000	14.24	42.7	45,622,200
1,000		200	168,000	1,000		200	168,000	1,000		200	168,000
3,300	14.24	42.7	3,860,340	1,700	14.24	42.7	1,988,660	2,100	14.24	42.7	2,456,580
			1,148,154				1,508,460				1,360,800
73,400			86,348,224	56,000			66,318,700	56,700			67,314,460
16,200		50	291,600	12,400		57	312,000	19,100		57	477,500
11,100	17.5	99	13,323,330	8,700	17.5	109	10,483,500	14,600	17	127	17,706,880
612,000		83	31,212,600	495,000		98.8	31,581,000	501,000		79	22,547,000
639,300			44,826,930	516,100			42,374,500	534,700			41,731,380
612,000		101.5	42,534,000	495,000		112	39,600,000	501,000		107	37,575,000
27,300		112	2,184,000	21,100		139	2,257,700	31,000		124	2,883,000
639,300			44,718,000	516,100			41,857,700	532,000			40,457,500
0			0	0			0	2,700		124	141,800
639,300			44,718,000	516,100			41,857,700	534,700			40,673,500
612,000		101.5	42,534,000	495,000		112	39,600,000	501,000		107	37,575,000
100		140	10,800	4,300		140	464,400	0			0
27,300		112	2,184,000	21,100		139	2,257,700	31,000		124	2,883,000
3,000		200	504,000	3,000		200	504,000	3,000		200	504,000
3,000		200	504,000	3,000		200	504,000	3,000		200	504,000
645,400		102.87	45,736,800	526,400		114.31	43,330,100	538,000		109.02	41,425,000
590,400		102.87	41,839,180	516,400		114.31	42,507,000	488,000		109.02	37,584,159
10,000		102.87	708,658	10,000		114.31	825,100	10,000		109.02	770,168
45,000		102.87	3,188,262	0			0	40,000		109.02	3,026,273
645,400			45,736,800	526,400			43,330,100	538,000			41,425,000
590,400		102.87	41,839,180	516,400		114.31	42,507,000	488,000		109.02	37,584,159
58,000	14.24	42.7	67,848,400	44,600	14.24	42.7	52,173,080	39,000	14.24	42.7	45,622,200
648,400		201.17	109,687,580	561,000		200.77	94,680,180	527,000		169.88	83,206,359
1,600		200	268,800	3,000		200	504,000	3,000		200	504,000
1,000		200	168,000	1,000		200	168,000	1,000		200	168,000
648,400		201.17	109,687,580	561,000		200.77	94,680,180	527,000		169.88	83,206,359
651,000		201.16	110,124,380	565,000		200.76	95,352,080	531,000		169.96	83,837,759
500		201.16	84,580	500		200.76	84,380	500		169.96	83,380
650,600		201.16	110,039,800	564,800		200.76	95,267,700	530,500		169.96	83,749,379
500				500				500			
3,000				3,000				3,000			

throughout the system enters the flow at this point as well as returns from the main unit glands and other miscellaneous clean-water returns with the exception of the high- and low-pressure drips. The water from this tank flows to the barometric condenser whose steam supply originates as described below. A sufficient quantity of steam is admitted to this condenser, the quantity being controlled as described in the preceding paragraph, to raise the water temperature to that desired for boiler feed.

7 Fig. 3 shows the position of the head tank, the barometric condenser and the boiler-feed tank, as well as the relative position of all steam apparatus in the heat-balance system.

8 Boiler-feed pumps and forced-draft fans, which are steam-driven, exhaust into an auxiliary exhaust header from which the evaporator is supplied. It is seldom, if ever, possible to make the quantity of low-pressure steam in this header exactly right for the evaporator demand. Consequently, there is a connection through a flow or back-pressure valve to the house-turbine exhaust pipe. This valve permits any steam in excess of that required by the evaporator to flow into the house-turbine exhaust pipe and thence to the barometric condenser. It is occasionally necessary to regulate the amount of water going to the barometric condenser, and this regulation is accomplished by means of a motor-operated valve controlled from the turbine-room floor. There is in conjunction with this valve a position indicator so that the operator on the turbine-room floor can tell just how far this valve is open.

9 The manipulation of this system is very simple and positive. For days at a time the temperature in the boiler-feed tank will not vary more than one or two degrees from 210 deg. fahr. This variation is, of course, due to fluctuations in the total station auxiliary load, which are, of course, reflected in the house-turbine load. A series of experiments was recently conducted to determine how closely this system could be regulated and what the lag might be between the time that the load on the house turbine was changed and the time when the effect was noticed in the feedwater tank. It was found that any degree of regulation was possible. The house-turbine load could be varied in steps of 10 kw., it being impossible to read variations less than this. The feedwater temperature could be varied by amounts as low as 0.5 deg. fahr., which is as close as an ordinary thermometer can be read.

10 Table 1 gives actual operating figures for five days, chosen so as to show the operating conditions over a range of loads from

30,000 to 60,000 kw. Data relating to these days are given in Table 2. The distribution of the auxiliary load is shown in Table 3.

TABLE 2 LOAD CONDITIONS DURING TEST

Test day	*A*	*B*	*C*	*D*	*E*
Time covered, hr.	4	24	24	24	24
Station load, kw.	61,750	50,290	50,000	40,460	38,790
Main unit load, kw.	61,850	50,615	50,440	41,660	39,500
House turbine load, kw.	1,150	1,110	1,110	180	470
Total elec. aux. load, kw.	1,250	1,435	1,580	1,380	1,180
Fans running, No.	2	2	2	2	2
Boiler-feed pumps running, No.	2	1	1	1	1
Average vacuum, main cond., in.	28.85	28.94	28.73	28.21	28.85

11 The figures given in these tables are presented exactly as they were worked out, without change or adjustment. All es-

TABLE 3 DISTRIBUTION OF AUXILIARY LOAD

Station load, kw.	60,100	39,500	50,600	50,400	41,600
House-turbine output, kw.	1,120	470	1,110	1,140	180
House transformers, kw.	140	710	375	375	1,195
Total aux. load, kw.	1,260	1,180	1,485	1,515	1,375
Input to stoker M. G. sets, kw.	56	52	56	56	45
" " lighting transf., kw.	15	32	36	38	36
" " 440-volt power, kw.	152	190	240	250	165
" " circulating pumps, kw.	600	475	660	690	700
" " air pumps, kw.	129	125	130	130	130
" " hot-well pumps, kw.	102	100	110	100	100
" " air compressors, kw.	56	56	56	56	56
" " exciters, kw.	150	150	197	195	143

timated quantities, that is, those that were not actually measured, are marked with an asterisk (*). These estimated quantities are not important and many of them could be, and were, accurately computed by the thermal equations of the various pieces of apparatus. It is believed that the balances shown are as nearly true as it is possible to make them. The data given are not considered representative of the best performance that can be expected from this station, but it is the intention to present in this paper only what has been actually accomplished.

12 The quantities shown in Fig. 2 are taken from the columns under "C," Table 1.

No. 1803 *c*

HEAT-BALANCE SYSTEM FOR HELL GATE STATION[1]

By J. H. Lawrence and W. M. Keenan, New York, N. Y.
Members of the Society

THE Hell Gate Station of The United Electric Light and Power Company, now being built in New York City, will have an ultimate capacity of 280,000 kw., consisting of eight 35,000-kw. turbo-generators. The initial generating installation will consist of two 25-cycle General Electric single-cylinder units and two 60-cycle Westinghouse double-cylinder units. The ultimate boiler installation will consist of twenty-four water-tube boilers, each having 18,900 sq. ft. of heating surface with double stokers equipped with clinker grinders. At present one-half the total number of boilers are being installed.

2 The station will be operated as four independent groups, each consisting of one 60-cycle, one 25-cycle unit, one house alternator and six boilers, although provisions have been made for interconnecting the various groups electrically and mechanically.

3 In connection with the design of this station, the following systems for driving the auxiliaries were considered:

a All steam
b Steam and direct current
c Steam and alternating current
d Direct current
e Alternating current.

After a careful study had been completed, it was decided to adopt the alternating-current system.

4 All auxiliaries in this station are to be motor-driven, with the exception of one boiler-feed pump for each group, one fire pump and one of two exciters for each house alternator. One-half the power for the auxiliaries of each group of two main units will be obtained from the main 60-cycle bus and one-half from the house alternator, so that in case of interruption of either source of power only one-half the auxiliaries will be shut down.

5 The house alternators will be 2500-kva. 60-cycle 3-phase 2300-volt direct-connected non-condensing machines, running at 3600 r.p.m. The power for the auxiliaries from the main station bus

[1] For discussion and closure see p. 516.

will be transformed from 13,200 to 2300 volts by means of two 7500-kw. power banks located in the boiler-house basement. All motors 25 hp. and over are to be operated at 2300 volts, excepting the motors for the coal towers, cranes and skip hoist, while motors under 25 hp. will be wound for 220 volts.

6 Any of the station auxiliaries, excepting the motors on the coal towers, cranes and skip hoist, may be operated from either the bus supplied by the main units or the house-alternator bus through the 2300-volt truck switches, which are located along the division wall between the turbine room and boiler-house basements. The house alternator and the bus supplied by the main units cannot be interconnected.

7 A diagram of the feedwater system of one group appears in Fig. 1. The condensate after leaving the condensers goes through closed heaters where it is heated to about 150 deg. fahr. and then enters the open heater where it is raised to 210 deg. fahr. The condensate is then pumped by the boiler-feed pumps direct to the boilers. Thermostatically controlled valves equalize the load on the two closed heaters. The closed heaters are supplied with steam at about 6.5 lb. absolute pressure bled from the lower stages of the main turbines. The condensate from the closed heaters is pumped to the open heater. As indicated in Fig. 1 fresh water obtained from the surge tank is used for the oil coolers and air-ejector condensers and discharges to the inlet of the closed heaters in order to conserve the heat from those different sources.

8 The house alternator exhausts at 1 lb. back pressure direct to the open heater. The exhaust from the high-pressure packing of the General Electric unit, approximately 4000 lb. of steam per hour, also flows directly into the open heater.

9 In order to insure a sufficient supply of steam to the open heaters at all loads on the house alternator, steam is bled from intermediate stages of the main turbines through pressure-control valves so adjusted as to automatically maintain 1 lb. back pressure in the open heaters.

10 The auxiliary control-board, by which one man will supervise all the station auxiliaries, is located above the turbine-room floor. Practically the only manual regulation required for securing the proper heat balance will be the control of the forced-draft fans which will be operated by push-button switches on the control board. The induced-draft fans after being manually started are regulated by an automatic balanced-draft system.

11 The motors for driving the forced- and induced-draft fans will be of the brush-shifting variable-speed type. The boiler-feed pumps will be driven by variable-speed wound-rotor induction

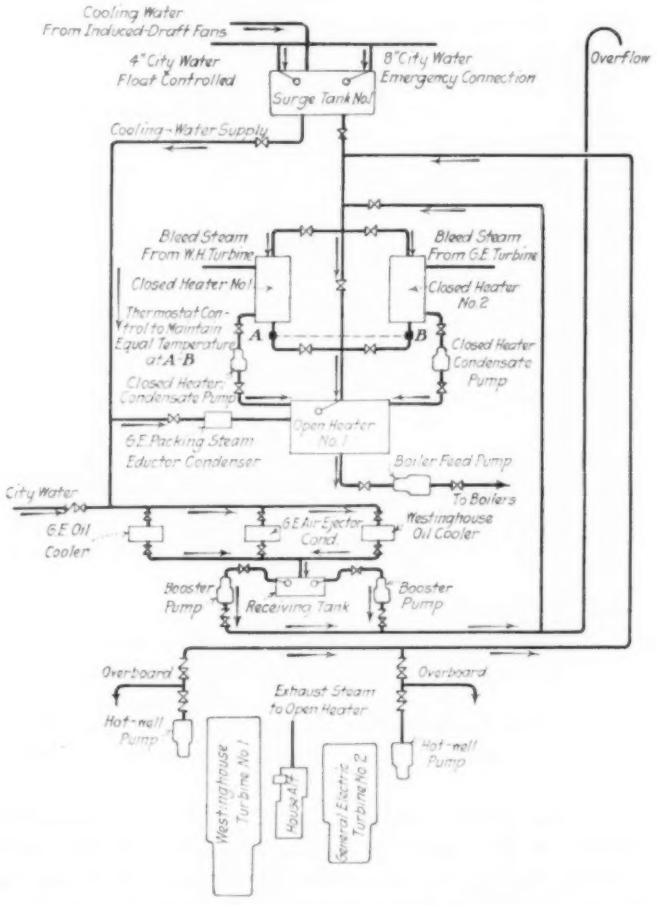


FIG. 1 HEATER CONNECTIONS FOR ONE SECTION

motors, while the remaining auxiliaries, such as the circulating-water, hot-well, dry-vacuum, booster and ash-handling pumps, will be driven by constant-speed squirrel-cage induction motors. The stokers will be driven by constant-speed squirrel-cage induction motors through variable-speed-transmission machines. The clinker grinders will be driven by two-speed squirrel-cage induction motors.

12 Fig. 2 shows a typical layout of panel on the control board. The instruments for each of the groups into which the station is divided are arranged as follows:

13 The draft gages in the top row are of the three-point type and indicate the furnace draft in each of the three boilers in each row of boilers.

14 The other instruments in the top row record: (*a*) The exhaust-steam temperature in the atmospheric exhaust line, in order

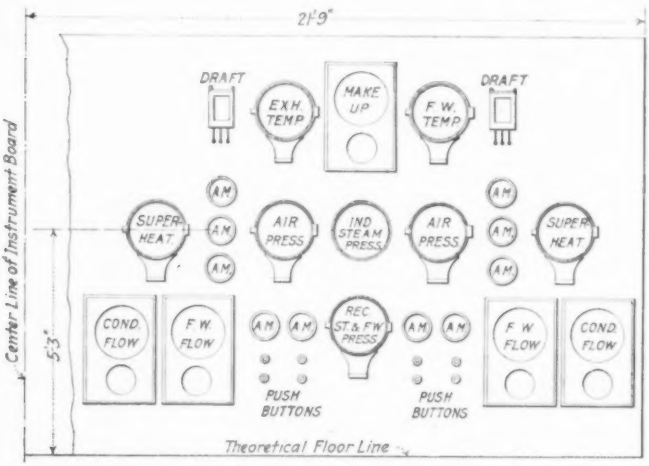


FIG. 2 PORTION OF INSTRUMENT BOARD FOR MECHANICAL APPARATUS, SHOWING LAYOUT FOR ONE OF THE FOUR GROUPS

to detect any loss due to excess exhaust steam; (*b*) the city water make-up to the group; (*c*) the temperature of the feedwater to the boilers.

15 In the middle row are: (*a*) Recorders for measuring the temperature of the steam at each of the main turbine throttles; (*b*) recorders showing the air pressure in the forced-draft duct leading to each row of boilers; (*c*) a steam-pressure gage for indicating the pressure in the main steam header; (*d*) ammeters for indicating the relative load on each of the induced-draft fans.

16 In the bottom row are recorders for measuring: (*a*) The condensate from each of the condensers; (*b*) the feedwater to each row of boilers; (*c*) by one instrument the main steam pressure and the feedwater pressure; (*d*) ammeters for indicating the relative load on the forced-draft fans.

17 It was the original intention to present herein the calculations made when computing the heat balance to be obtained, but after conferring with the various authors who were preparing papers on heat balance for the Meeting, it was felt that the chief interest would be centered upon operating figures, and as the Hell Gate Station is not in operation at the time this paper goes to press, operating figures are not available, and the heat balance calculated for it would therefore have only theoretical value.

18 It was thought, however, that the presentation of the system to be employed to obtain the heat balance would be of interest at this time, and for this reason the present brief description was prepared. It is to be hoped that operating figures will be available for discussion at the Annual Meeting.

No. 1803 *d*

HEAT BALANCE OF THE CONNORS CREEK PLANT OF THE DETROIT EDISON COMPANY

By C. Harold Berry,[1] Detroit Mich.
Member of the Society

and

F. E. Moreton,[1] Detroit, Mich.
Non-Member

THE term "heat balance" is currently used in two different senses. At times we speak of the heat balance when we mean a physical condition in the plant whereby there is a balance of certain heat-absorbing and heat-developing capacities. Again, by heat balance we mean a thermal balance sheet which records the ultimate disposition of all the heat developed from the fuel used.

2 This paper discusses the heat balance in both senses for the Connors Creek Plant of The Detroit Edison Company. We shall first describe briefly the apparatus in the plant, following this by a discussion of the ideal operating conditions, and finally presenting actual results. The plant is considered just as it stands, without reference to changes now under consideration or construction.

DESCRIPTION OF EQUIPMENT

3 The main turbo-generators are six in number — three[2] of 20,000 kw. capacity, one of 30,000 kw. capacity, and two of 45,000 kw. capacity, giving a total installed capacity of 180,000 kw.[3] The turbines are served by fourteen boilers, each of 2365 boiler hp. builder's nominal rating.

4 The boiler-feed pumps are steam-turbine-driven, with the exception of one which is motor-driven. One general service-water pump is steam-turbine-driven. All other plant auxiliaries are motor-driven, some by alternating-current motors, some by direct-current motors, the choice depending upon the nature of the driven

[1] The Detroit Edison Company.

[2] At the present writing two of these are in the manufacturer's plant in the course of rebuilding. A 10,000-kw. unit stands in place of one of them.

[3] For the time being 150,000 kw.

unit. As it works out, those auxiliaries whose uninterrupted running is essential to plant operation are all driven by direct-current motors, while auxiliaries whose stoppage does not immediately affect the rest of the plant are driven by alternating-current motors. This division is not due to any belief in the superior reliability of direct-current motors, but is due to the fact that most, if not all, of the essential auxiliaries are of such character as to require or benefit

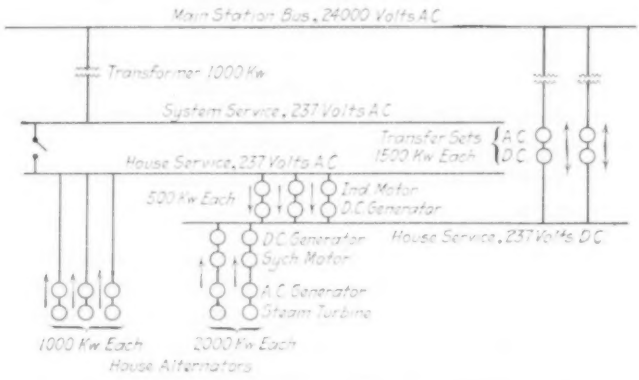


FIG. 1 ARRANGEMENT OF AUXILIARY ELECTRICAL SYSTEM, CONNORS CREEK PLANT

from widely variable speed. Such are stokers, blowers, hot-well pumps,[1] dry-vacuum pumps, circulating pumps, etc.

5 The arrangement of the auxiliary electrical system is shown in Fig. 1. Alternating current for auxiliaries is available from either of two buses: (*a*) The "system service" bus, which is fed through a transformer from the main station bus, or (*b*) the "house service" bus, which is fed by three 1000-kw. turbo-generators. A tie switch is provided whereby these two buses can be connected. When this tie switch is closed, such of the 1000-kw. house alternators as are running must perforce operate in parallel with the main units. When this is done (and it is common practice at Connors Creek), the governors of the small units are set for a speed slightly above system frequency, and the house units are operated on the hand throttle, with constant steam flow and virtually constant load.

6 Direct current is available from a single bus (actually con-

[1] At the time of writing, the older of these pumps still have induction motors, which are to be displaced by direct-current motors.

structed as a ring). Owing to the importance of continuous service from the motors connected to this bus, it is fed from three sources:

a Two 1500-kw. steam-driven direct-current generators. Each of these consists of a turbo-alternator permanently connected (by bolted links) to a synchronous motor-direct-current generator set, so that the combination is merely a steam-driven direct-current generator with an electrical speed reduction instead of gears.

b Three 500-kw. motor-generator sets (induction motors) driven from the house service alternating-current bus.

c Two 1500-kw. motor-generator sets (synchronous motors) driven from the main station bus. These sets have characteristics and controls such that they may deliver power in either direction, wherefore they are known as "transfer sets."

7 An important characteristic of a system of this sort is that the load on the auxiliary steam turbines is independent of the auxiliary power demand of the station. With the tie switch closed between the system service and house-service buses, and with the transfer sets operating between the main system and the house-service direct-current bus, the house-turbine loads may be adjusted at will, within limits set by the capacities of the apparatus.

8 The function of the house turbines is twofold:

a To furnish a standby source of power for station auxiliaries in the event of a system failure, and

b To furnish exhaust steam for boiler feedwater heating.

9 Fig. 2 shows the arrangement of the auxiliary exhaust-steam and feedwater heating apparatus. The auxiliary exhaust header receives the exhaust steam from five house turbines, from six boiler-feed-pump turbines, and from one general-service-pump turbine, as well as the vapor formed in the boiler-feed make-up evaporators. This steam passes upward into five barometric-condenser heads, where it meets condensate from the main condensers. The resulting hot water is discharged into the hot boiler-feed tanks, whence it is pumped into the boilers. The cold condensate piping and tanks are designed to care for variations in the volume of water in the boilers at times of variable plant load, and to safeguard the boiler-feed pumps against failure of the supply. Inasmuch as these barometric condensers serve as condensers for the auxiliary system

and as feedwater heaters for the main system, they are known as "heater condensers."

THEORY OF OPERATION

10 There are two viewpoints from which to state the gain in station economy due to the operation of a system of this kind:

a A simple but approximate view — the substitution of cheap power for costly power

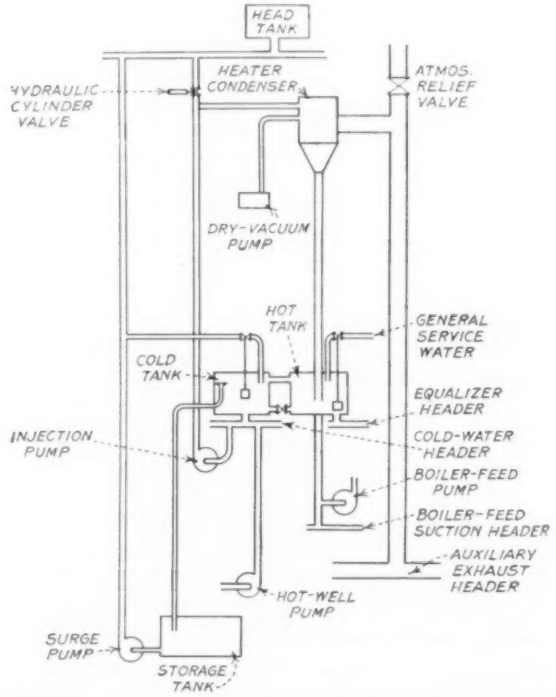


FIG. 2 ARRANGEMENT OF AUXILIARY EXHAUST-STEAM AND FEEDWATER HEATING APPARATUS, CONNORS CREEK PLANT

b A complicated but exact computation — a saving due to increased feedwater temperature, offset by an increased station steam consumption.

The matter will be discussed from each of these two points of view.

11 In the chart shown in Fig. 3 the full lines show the com-

puted cost of power for variable turbine steam consumption and boiler-feed temperature, with fixed initial steam conditions. If we assume that all losses in the turbine appear as heat available from the exhaust steam, that is, if we neglect radiation losses, and if we

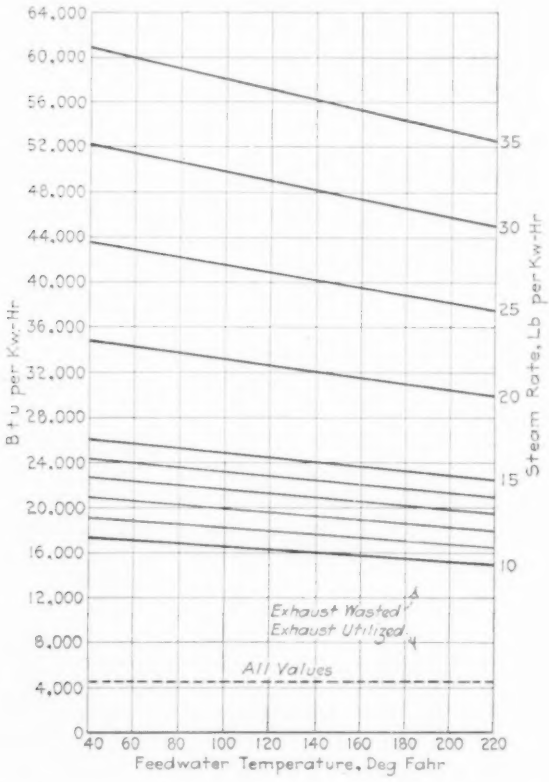


FIG. 3 COMPUTED COST OF POWER FOR VARIABLE TURBINE STEAM CONSUMPTION AND BOILER-FEED TEMPERATURE, WITH FIXED INITIAL STEAM CONDITIONS

(Steam pressure, 240 lb. per sq. in. abs.; superheat, 200 deg. fahr.; boiler-room efficiency, 0.75)

recover usefully and credit to the turbine all of the heat available from the exhaust steam, then the cost of power is constant for all conditions, and is equal to the heat equivalent of a kilowatt-hour (3415 B.t.u.) corrected only for boiler-room losses. For the assumed

boiler-room efficiency of 0.75, this gives us a cost of power of 4550 B.t.u. per kw-hr., and this is shown plotted in Fig. 3 as a dashed line.

12 In the case of a plant like Connors Creek, the main units have a steam consumption of, let us say, 12 lb. per kw-hr., wherefore power generated by them costs from 18,000 to 21,000 B.t.u. per kw-hr. The house turbine, whose exhaust heat we may assume to be fully utilized, produces power at a cost of 4550 B.t.u. per kw-hr. Clearly, the displacement of a portion of the power generated by the main

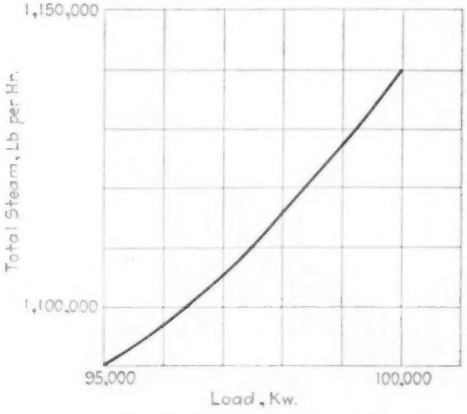


FIG. 4 TOTAL-STEAM CURVE OF MAIN UNIT

units at high cost by power generated by the house turbine at much lower cost will result in a gain in station economy. From this it would appear that the maximum house-turbine output is to be striven for. But the matter is not so simple, for here we meet the outstanding feature of the system — the auxiliary exhaust pressure may be adjusted to any value between the lowest attainable absolute pressure and atmospheric pressure. If we operate at a very low pressure, the auxiliary exhaust steam, and therefore the boiler feedwater, are at low temperature, with a resulting high cost of power from the main units. Further, with a small temperature rise of the boiler feedwater, very little exhaust steam can be condensed, wherefore the output of the house turbine is small. As the auxiliary exhaust pressure rises, the boiler-feed temperature rises, lowering the cost of power from the main units, and at the same time the heat absorption of the boiler feedwater increases, condensing more auxiliary

exhaust steam and permitting the generation of more cheap power by the house turbine. As the auxiliary exhaust pressure continues to rise, however, another influence enters: the steam consumption of the house turbine increases, that is, the power output of a given steam flow will be less. This reduces the proportion of cheap power generated, and eventually a point is reached at which an increase of auxiliary exhaust pressure will decrease the station economy.

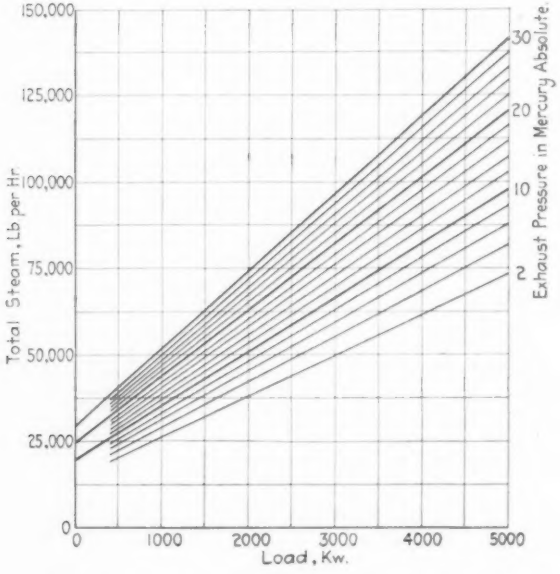


FIG. 5 TOTAL-STEAM CURVES OF HOUSE TURBINE

Our problem is to locate this point of maximum station economy. It will be worked out in the more exact analysis which follows.

13 As is pointed out below, in discussing the results of this study, a complete analysis of this problem requires a large fund of information, which unfortunately is not yet available for the Connors Creek plant. We shall therefore carry out this study for an assumed plant with characteristics consistent with the performance of Connors Creek. The results obtained will serve to define the problem rather than to present a solution for the actual plant. The following data are assumed:

Total station load, including auxiliary power demands, constant and equal to 100,000 kw.

Main unit steam consumption assumed to conform to the total-steam curve of Fig. 4

A house turbine of 5000 kw. capacity, whose performance is shown by the total-steam curves of Fig. 5 [1]

Initial specific total heat of steam at 225 lb. per sq. in. gage and 200 deg. superheat = 1313 B.t.u. per lb.

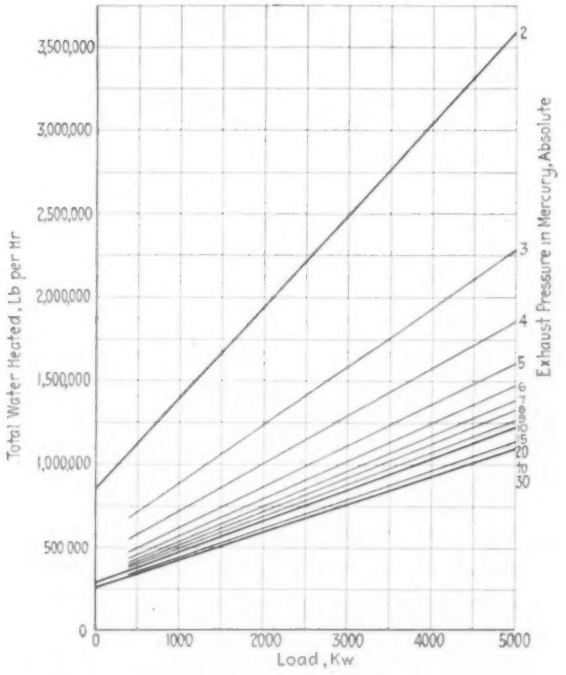


FIG. 6 WATER HEATABLE BY EXHAUST STEAM FROM HOUSE TURBINE

Temperature of condensate leaving main unit condensers = 70 deg. fahr.

Boiler feedwater leaves the heater condensers at a temperature 10 deg. fahr. below the boiling point corresponding to the auxiliary exhaust pressure

Boiler-room efficiency equal to 0.75

Radiation neglected.

[1] These curves are based on an actual test of a 1000-kw. set, and probably indicate a higher steam consumption than is normal for a 5000-kw. unit.

14 Our first step is to develop a very interesting relation which applies to any steam prime mover whose exhaust is wholly used for heating water. If the water is heated from a constant initial temperature to a constant number of degrees below the boiling point corresponding to the exhaust pressure, then a curve plotted between

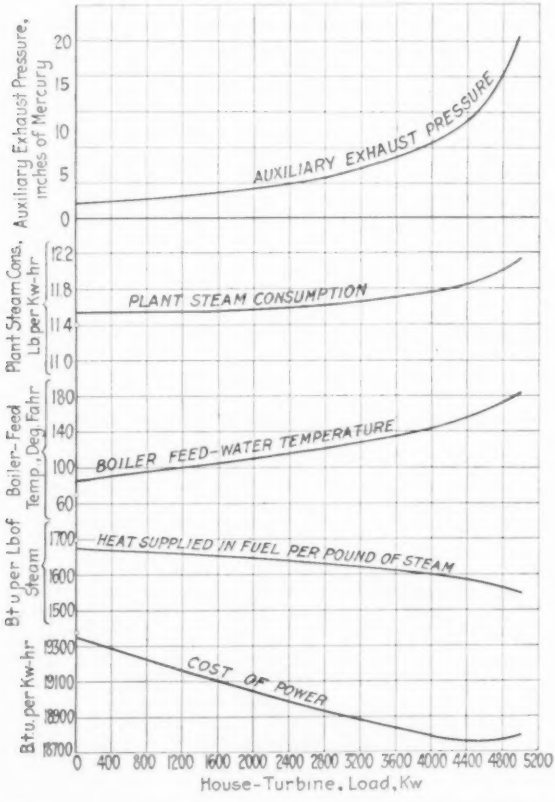


FIG. 7 COST OF POWER FOR VARIABLE HOUSE-TURBINE LOAD

the turbine load and the quantity of water capable of being heated will have the same shape as the curve plotted between turbine load and total steam. Appendix No. 1 gives the proof of this fact, and develops the relations involved for the conditions assumed above. For these conditions the curves of Fig. 6 have been prepared.

15 We are now in a position to determine the cost of power for variable house-turbine load. Appendix No. 2 indicates the

method and gives the computations in detail. The principal results are shown in Figs. 7 and 8. From the curves it is clear that the best station economy is obtained with a house-turbine load of 4500 kw. and a boiler-feed temperature of 160 deg. fahr. In Fig. 7 the small circles at zero house-turbine load indicate conditions with the house turbine shut down.

16 In interpreting these results, attention must be directed to the fact that the curve of cost of power is rather flat. Operation at

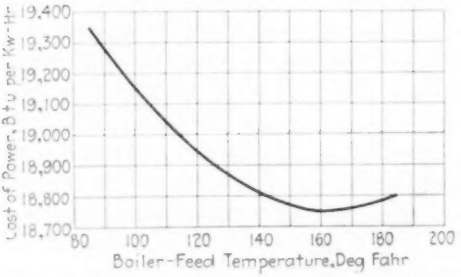


FIG. 8 VARIATION OF COST OF POWER WITH BOILER-FEED TEMPERATURE

a condition somewhat removed from the best point results in only a small increase in the cost of power. However, modern power-plant operation is largely concerned with small increments, and a gain of even 100 B.t.u. per kw-hr. is not to be neglected.

17 Attention must further be called to the fact that the computations here given are based on one set of assumed conditions. To carry out this study in general would require much detailed information. We should need the total-steam curve for the main units running in various combinations over the entire range of plant load. We should also need the total-steam curves for all house turbines (at Connors Creek, five house alternators, six boiler-feed pumps, one general-service pump, — twelve turbines in all) over their entire range of load and exhaust pressure. We should then have to develop combined curves for various load distributions among these auxiliaries, and for each of these we should have to compute results for various plant loads and main unit distributions, for various values of the condensate temperature and for various values of the interval d by which the boiler-feed temperature is below the boiling point corresponding to the auxiliary exhaust pressure. This is obviously a large undertaking, and one for which we have not yet secured adequate data.

18 The main facts, however, are clear. For a system of this type there is a best point, and it is just as wasteful to heat the feedwater to a higher temperature as to a lower. We believe that for the Connors Creek plant the best economy is obtained with a boiler-feed temperature lying between 160 and 180 deg. fahr., and in this range we try to operate.

BALANCE SHEET

19 The preparation of the accompanying balance sheet, Table 1, calls for the exercise of considerable judgment in selecting data, and even for some "directed guessing." The major items are, of course, taken from direct observations, but such things as radiation and drip losses are not susceptible of measurement by practical means, and must be estimated on the basis of other observed data or simply on general considerations. Appendix No. 3 outlines in full detail the method of securing each value on the sheet.

The fact that each item in the "Credit, Useful" column reappears in the "Debit" column, together with the fact that each item, as well as the entire sheet, must balance, introduces numerous cross checks between observed and estimated values, so that the table represents actual plant conditions fairly accurately. The actual boiler room efficiency of the plant for the month of April 1921, was 74.9 per cent, which checks within 0.2 per cent of the value shown in the table. The actual thermal efficiency of the plant was identical with the figure for "Energy sent out," as per cent of heat value of the coal. All of the values are averages for the month of April 1921, and are based on coal weights by track scales (with estimation of bunker contents at beginning and end), heating value by bomb calorimeter, water fed to boilers by Venturi meters, and electrical output by integrating meters.

APPENDIX NO 1

THE RELATION BETWEEN HOUSE-TURBINE LOAD AND THE QUANTITY OF WATER WHICH CAN BE HEATED FROM A FIXED INITIAL TEMPERATURE TO *d* DEGREES BELOW THE BOILING POINT CORRESPONDING TO THE AUXILIARY EXHAUST PRESSURE

20 *Item 1.* The decrease in the specific total heat of the steam in its passage through the house turbine is the sum of the work developed per pound

of steam (load in kw., multiplied by 3415, divided by the total pounds of steam per hour) plus the heat lost by radiation from the turbine per pound of steam:

$$\frac{3415\,L}{S} + R$$

21 *Item 2.* The specific total heat of the exhaust steam (H_x) equals the initial specific total heat of the live steam (H_i), minus the decrease in specific total heat of the steam as found in Item 1:

$$H_x = H_i - \frac{3415L}{S} - R$$

22 *Item 3.* The heat available from one pound of exhaust steam for heating water to d degrees below the boiling point (t_x) corresponding to the exhaust pressure equals the specific total heat of the exhaust steam, minus the specific total heat of liquid water at the temperature ($t_x - d$). With ample accuracy for the problem under consideration, this becomes

$$H_x - (t_x - d - 32) = H_i - \frac{3415L}{S} - R - t_x + d + 32$$

23 *Item 4.* The heat required to heat one pound of water from its initial temperature (t_w) to ($t_x - d$) is

$$t_x - d - t_w$$

24 *Item 5.* The pounds of water which can be heated by one pound of exhaust steam will then be

$$\text{Item 3} \div \text{Item 4} = \frac{H_i - \dfrac{3415L}{S} - R - t_x + d + 32}{t_x - d - t_w}$$

TABLE 2 CALCULATION OF WATER HEATABLE BY TURBINE EXHAUST STEAM

Exhaust pressure, in. Hg.	t_x deg. fahr.	$\frac{1355 - t_x}{t_x - 80}$	$\frac{3415}{t_x - 80}$	L = 1000 kw.				L = 5000 kw.			
				S	$\frac{1355 - t_x}{t_x - 80} S$	$\frac{3415}{t_x - 80} L$	W	S	$\frac{1355 - t_x}{t_x - 80}$	$\frac{3415}{t_x - 80} L$	W
2	101	59.71	162.6	26100	1558000	162600	1395000	73500	4389000	813000	3576000
3	115	35.42	97.5	27600	978000	97500	881000	78100	2767000	487500	2280000
4	125	27.33	75.9	29000	793000	75900	717000	81800	2237000	379500	1858000
5	134	22.60	63.2	30100	680000	63200	617000	85100	1923000	316000	1607000
6	141	19.90	56.0	31300	623000	56000	567000	88000	1751000	280000	1471000
7	147	18.03	50.9	32500	586000	50900	535000	90700	1636000	254500	1382000
8	152	16.72	47.4	33400	558000	47400	511000	93400	1562000	237000	1325000
9	157	15.56	44.3	34400	535000	44300	491000	95800	1490000	221500	1269000
10	162	14.55	41.6	35200	512000	41600	470000	98300	1430000	208000	1222000
15	179	11.88	34.5	39600	470000	34500	436000	110000	1307000	172500	1135000
20	192	10.38	30.5	43800	455000	30500	425000	120700	1253000	152500	1101000
25	203	9.36	27.8	47900	448000	27800	419000	131300	1228000	139000	1089000
30	212	8.66	25.9	51700	448000	25900	422000	141500	1226000	129500	1097000

TABLE 1 THERMAL BALANCE SHEET FOR APRIL 1921, THE DETROIT EDISON CO., CONNORS CREEK POWER HOUSE

The month of April, 1921, represents neither the best nor the current plant performance. At that time the plant was still operating with crippled units and with much equipment still in progress of recovery from post-war conditions.

Apparatus	Debit				Credit: Useful				Credit: Lost				Efficiency of Apparatus
		B.t.u. per lb. of coal	Per cent of heat value of coal	B.t.u. per kw-hr.		B.t.u. per lb. of coal	Per cent of heat value of coal	B.t.u. per kw-hr.		B.t.u. per lb. of coal	Per cent of heat value of coal	B.t.u. per kw-hr.	
Boiler	Coal	12349	100	20140	Steam to mains (sup.)	9875	79.97	16175	Moisture (hyd.)	560	4.53	914	$\frac{10454-1180}{12349} = 75.1\%$
	Main Alternator	59	0.48	96	Steam to mains (sat.)	579	4.69	945	Moisture in air	25	0.20	41	
	Air House Alternator	2	0.02	3					CO	65	0.53	106	
	Boiler Feed	1180	9.55	1927					Carbon in ref.	370	3.00	604	
	Radiation from Mains	33	0.27	54					Radiation	225	1.82	367	
	Work by B. F. Pumps	8	0.06	13					Dry gases	1932	15.65	3152	
		13631	110.38	22233		10454	84.66	17060		3177	25.73	5184	
Steam Mains	Steam from Boiler	10454	84.66	17060	Steam to turbines	9843	79.72	16070	Heating system	35	0.28	57	$\frac{9843+82}{10454} = \frac{9925}{10454} = 94.9\%$
					Steam to evaporators	82	0.66	134	Flue blowers	61	0.49	100	
					Radiation (mains)	33	0.27	54	Soot blowers	120	0.97	196	
					High press. Traps	5	0.04	8	Leaks (restaurant, etc.)	265	2.15	433	
					Heating system traps	10	0.08	16					
		10454	84.66	17060		9973	80.77	16282		481	3.89	786	
Turbines	From Steam Mains	9160	74.18	14960	Energy to shaft	2205	17.86	3600	Bearing friction	40	0.32	65	$\frac{2205}{9160-484} = \frac{2205}{8676} = 25.4\%$
					Packing loss	196	1.59	320	Radiation	10	0.08	16	
					To condenser	6709	54.33	10950					
		9160	74.18	14960		9110	73.78	14870		50	0.40	81	
Condenser	From Turbine	6709	54.33	10950	To condensate	484	392	790	To circ. water	6421	52.00	10480	
Condenser	From Turbine Packing	196	1.59	320									
		6905	55.92	11275		484	392	790		6421	52.00	10480	
Main Alternators	Energy from Shaft	2205	17.86	3600	To switchboard	2106	17.04	3437	Bearing friction	30	0.24	49	$\frac{2106}{2205} = 95.5\%$
					Ventilating air	59	0.48	96	Exciter loss	10	0.08	16	
		2205	17.86	3600		2165	17.52	3533		40	0.32	65	
House Turbine	Steam from Mains	568	4.60	927	Energy to shaft	51	0.41	83	Bearing friction	5	0.04	8	
					To heater-condensers	507	4.11	828	Radiation	1	0.01	2	
									Packing loss	4	0.03	6	
		568	4.60	927		558	4.52	911		10	0.08	16	

House Alternator	Energy from Shaft....	51	0.41	83	To switchboard.....	43	0.35	70	Bearing friction.....	5	0.04	8	$\frac{43}{51} = 84.3\%$
					Ventilating air.....	2	0.02	3	Exciter loss.....	1	0.01	2	
		51	0.41	83		45	0.37	73		6	0.05	10	
Heater Condenser	From House Turbine...	507	4.11	828	Feedwater.....	1185	9.60	1934	Radiation.....	5	0.04	8	
	From B.F.P. Turbine..	106	0.86	173									
	Condensate.....	484	3.92	790									
	Evaporator.....	78	0.63	127									
	Storage Tank.....	15	0.12	24									
		1190	9.64	1942		1185	9.60	1934		5	0.04	8	
Boiler-Feed Pump Turbines	Steam from Mains.....	115	0.93	188	Heater-condensers ...	106	0.86	173	Radiation.....	1	0.01	2	
					Work on feed water...	8	0.06	13					
		115	0.93	188		114	0.92	186		1	0.01	2	
Evaporator	Steam from Mains.....	82	0.66	134	To heater-condensers .	78	0.63	127					
	Heat in Raw Water....	1	0.1	2	To storage tank.....	5	0.04	8					
		83	0.67	136		83	0.67	135					
Boiler-Feed Tanks	From Feed Water.....	1185	9.60	1934	To boilers.....	1180	9.55	1927	Radiation.....	10	0.08	16	
	From High-Press. Traps	5	0.04	8									
		1190	9.64	1942		1180	9.55	1927		10	0.08	16	
High- and Low-Pressure Traps	From Steam Mains....	5	0.04	8	To B.F. tank	5	0.04	8					
	From Heating System .	10	0.08	16	To storage tank.....	10	0.08	16					
		15	0.12	24		15	0.12	24					
Storage Tanks	From Heating System..	10	0.08	16	To heater-condensers .	15	0.12	24					
	Evaporator.....	5	0.04	8									
		15	0.12	24		15	0.12	24					
Switchboard	From Main Alternators	2106	17.04	3437	Energy sent out.....	2093	16.95	3415	To stoker motors.....				Ratio Auxiliary to Net delivered kw-hr.
	From House Alternators	43	0.35	70					circulators.....				
									coal handling.....				
									blowers.....				
									compressor.....	56	0.45	91	$= \frac{656}{2093} = 2.68\%$
									D.V. pumps.....				
									transmission.....				
									miscellaneous.....				
									gen. service pumps..				
		2149	17.39	3507		2093	16.95	3415		56	0.45	91	
Totals		47731	386.52	77901		37474	303.50	61164		10257	83.05	16739	

25 *Item 6.* The total water heated, lb. per hr. (W), will be the product of Item 5, multiplied by the total steam, lb. per hr. (S):

$$W = \frac{\text{Item 3}}{\text{Item 4}} \times S$$

$$= \frac{H_i - \frac{3415L}{S} - R - t_x + d + 32}{t_x - d - t_w} \times S$$

$$= \left[\frac{H_i - R - t_x + d + 32}{t_x - d - t_w}\right] S - \left[\frac{3415}{t_x - d - t_w}\right] L$$

26 Since W is a linear function in S and L, the W–L curve will be of the same class as the S–L curve. The S–L curve being known, for any exhaust pressure (t_x thus being fixed), for known or assumed values of d, t_w, and R, the computation of the W–L curve is simple and direct.

27 The following values are here assumed:

$$H_i = 1313 \qquad d = 10$$
$$t_w = 70 \qquad R = 0$$

28 Substituting these in the formula developed above gives, for the special problem under consideration,

$$W = \left[\frac{1355 - t_x}{t_x - 80}\right] S - \left[\frac{3415}{t_x - 80}\right] L$$

29 In calculating W in the case assumed, advantage is taken of the fact that the total-steam curves of Fig. 5 were assumed to be straight lines, wherefore the W-curves will also be straight lines. Accordingly, two points are computed for each line, one for a load of 1000 kw. and one for a load of 5000 kw. In the tabulation which follows attention is called to the fact that W decreases with increasing exhaust pressure up to about 25 in. of mercury, after which W increases slightly with increasing exhaust pressure. Between 20 and 30 in. of mercury, however, the difference in W is too small to plot clearly. For this reason a single line is drawn for this entire range in Fig. 6, which embodies the results computed in Table 2.

APPENDIX NO. 2

COMPUTATION OF THE COST OF POWER FOR VARIABLE HOUSE-TURBINE LOAD

30 Having developed the data set forth in Figs. 5 and 6, proceed as follows:

31 *Step 1.* Assume a house-turbine load.

32 *Step 2.* Find the main unit load by subtracting the assumed house-turbine load from the total plant load (in this case 100,000 kw.).

33 *Step 3.* From the total-steam curve for the main units, find the total steam per hour corresponding to the main unit load as found in Step 2. This is also the weight of condensate to be heated by the auxiliary exhaust steam.

34 *Step 4.* From Fig. 6, determine the auxiliary exhaust pressure at which the house-turbine load assumed in Step 1 will correspond with the heating of the water computed in Step 3.

35 *Step 5.* For the assumed house-turbine load and the auxiliary exhaust pressure determined in Step 4, from Fig. 5 determine the total steam delivered to the house turbine per hour.

36 *Step 6.* Find the total steam for the entire plant by adding the main and auxiliary steam, as found in Steps 3 and 5.

37 *Step 7.* Find the plant steam consumption by dividing the total steam as found in Step 6 by the plant load (100,000 kw.).

38 *Step 8.* Determine the temperature of the boiler feedwater, which is d (in this case, 10) degrees below the boiling point corresponding to the auxiliary exhaust pressure as found in Step 4.

39 *Step 9.* Determine the heat required in fuel to produce one pound of steam. This equals the initial specific total heat of live steam (H_i = 1313 B.t.u. per lb.), minus the specific total heat of boiler feedwater (its temperature, Step 8, minus 32), divided by the boiler-room efficiency (in this case assumed as 0.75).

40 *Step 10.* Determine the cost of power in B.t.u. per kw-hr. by multiplying the B.t.u. required to produce one pound of steam (Step 9) by the plant steam consumption (Step 7).

41 Repeat this entire procedure for various house-turbine loads. For the conditions assumed in this paper the detailed computations are given in Table 3. This table comprises only a few of the points computed, and the values given have been rounded up by reading back from curves plotted to the computed values. The principal results are shown in Figs. 7 and 8.

TABLE 3 CALCULATION OF THE COST OF POWER FOR VARIOUS HOUSE-TURBINE LOADS

House turbine load, kw.	Main unit load, kw.	Total steam to main units, lb. per hr.	Auxiliary exhaust pressure, in. Hg.	Total steam to house turbine, lb. per hr.	Total steam to plant, lb. per hr.	Plant steam consumption, lb. per kw-hr.	Boiler feed temp., deg. fahr.	Heat supplied in fuel per lb. steam, B.t.u. per lb.	Cost of power, B.t.u. per kw-hr.
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)
HOUSE	TURBINE	SHUT	DOWN:						
0	100000	1140000	.....	0	1140000	11.40	70	1700	19380
HOUSE	TURBINE	RUNNING							
0	100000	1140000	1.7	14200	1154200	11.54	86	1677	19350
1000	99000	1127500	2.3	27000	1154500	11.55	96	1663	19190
2000	98000	1116000	3.4	41000	1157000	11.57	110	1646	19040
3000	97000	1106000	5.1	57800	1163800	11.64	124	1625	18910
4000	96000	1097000	8.5	79500	1176500	11.77	144	1598	18790
4500	95500	1093500	11.9	94300	1187800	11.87	160	1580	18750
5000	95000	1090000	21.0	123000	1213000	12.13	185	1548	18790

DISCUSSION

JOHN A. STEVENS. In friendly criticism of the steam engineering of the stations described in these papers, the writer would criticize the steam pressures and temperatures used in each case, especially as they were all engineered by our largest engineering firms, and because some of these stations might be considered a part of the Eastern Superpower Zone, and at some subsequent date may be a part of generating stations of the so-called Superpower System. If they serve their purpose, these stations will have to furnish power to the majority of consumers at something less than the consumers

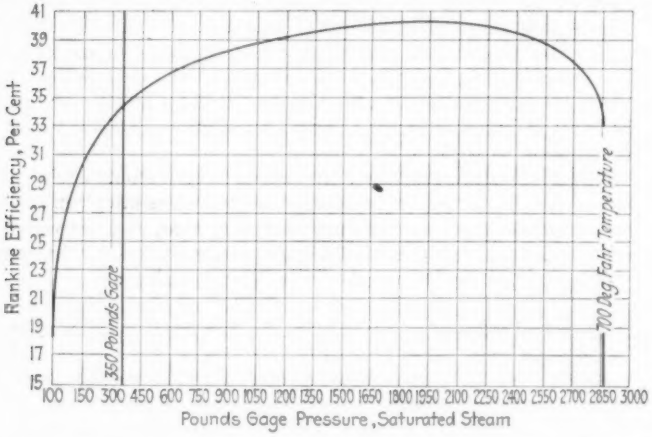


FIG. 1 RANKINE EFFICIENCY FOR VARIOUS INITIAL PRESSURES, DRY STREAM, BACK PRESSURE, 1 IN. HG., FINAL TEMPERATURE 79 DEG. FAHR.

can make it themselves. The exact relative difference in the cost of production, the writer is not prepared to quote, and he does not believe anybody could at this time.

The Superpower Survey [1] recommends a steam pressure at the throttle of 300 lb. per sq. in., which would mean 325 or 350 lb. at the boilers, and a superheat at the throttle of 230 deg. fahr., which at the boilers would have to be 240 to 250 deg., making a temperature at the throttle of 652 deg. fahr., or at least 686 deg. at the boilers. An absolute pressure at the exhaust nozzle of 1 in. of mercury is also recommended. The engineers who made up these recommendations, in all probability, propose these pressures and superheats for the turbine side of the throttle or at the first-stage nozzles.

[1] Technical Paper No. 123, U.S. Geol. Survey, p. 23.

The Prime Movers Committee's Report of the National Electric Light Association quotes a list of high-pressure and high-temperature stations in this country and Europe found in Table 1.

TABLE 1 DATA ON HIGH-PRESSURE AND HIGH-TEMPERATURE POWER PLANTS

Name of plant	Name of company	Gage pressure, lb. per sq. in.	Superheat, deg. fahr.	Total temp., deg. fahr.
River Plant	Buffalo General Elec. Co.	275		675
Joliet, Ill.	Public Service Co. of Northern Illinois	325		700
Windsor	American Gas & Elec. Co.	250		675
West Penn	Union Gas & Elec. Co.	250		640
Springdale	West Penn Power Co.	350		670
Colfax	Cheswick Power Co.			
	at boilers	275	200	600
	at throttle	265	175	586
Calumet	Commonwealth Edison Company	300		650
Marysville	Detroit Edison Co.	300		620
Extension to L Street	Boston Edison Co.	300		620
South Meadow	Hartford Elec. Light Co.	275		640
Hell Gate	United Elec. Light Co.	275	186 (approx)	606
Colliery in South Wales		350		
B.T.H.	Rugby, England	350		700
U.D.E.	Paris, France	355		700
Teesbank, North Tees.	Newcastle Supply Co.	475*		720
Connors Creek	Detroit Edison Co.	225	200†	597
Delaware	Philadelphia Elec. Co.			
	at boiler	250	180	586
	at throttle	235	174	576
Northeast	Kansas City	300†	250	675
Proposed by General Electric Co. for Paris Station		350	350	786

* N. E. L. A. figure is 450.
† At boilers.

It will be noted that the stations discussed in the present papers are designed for comparatively low pressure and low temperature as compared with many other stations in this country as well as the more recent stations in Europe which are quite in advance of anything in American practice of the present day regarding pressure and temperature.

The steam curves of Figs. 1, 2 and 3 indicate the relative efficiency at different pressures and superheats. In the writer's judgment, all large central stations constructed from now on should

be built for a working pressure at the boilers of at least 350 lb. per sq. in. and a total temperature at the boilers of at least 700 deg. fahr.

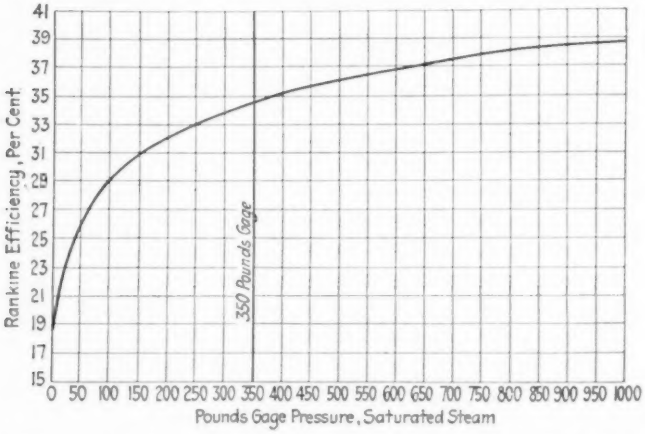


FIG. 2 RANKINE EFFICIENCY FOR VARIOUS INITIAL PRESSURES, DRY STREAM, BACK PRESSURE, 1 IN. HG., FINAL TEMPERATURE 79 DEG. FAHR.

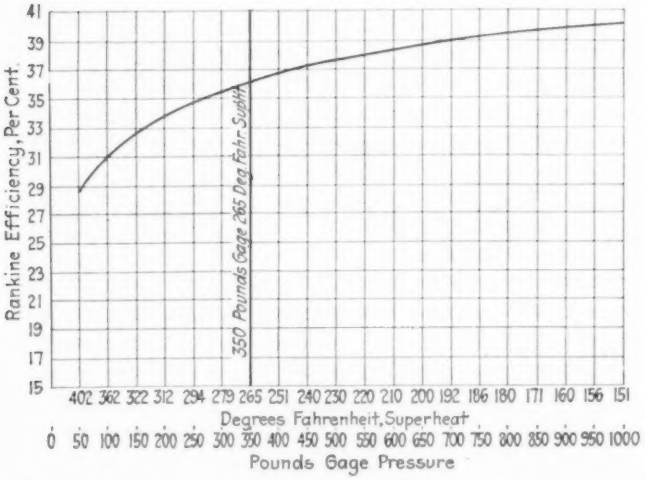


FIG. 3 RANKINE EFFICIENCY FOR VARIOUS INITIAL PRESSURES WITH CONSTANT INITIAL TEMPERATURE 700 DEG. FAHR., BACK PRESSURE, 1 IN. HG., FINAL TEMPERATURE 79 DEG. FAHR.

The piping should be so constructed and insulated that there would be at least 300 lb. pressure and at least 230 deg. superheat at the first-stage nozzles, which would give a total temperature of at least

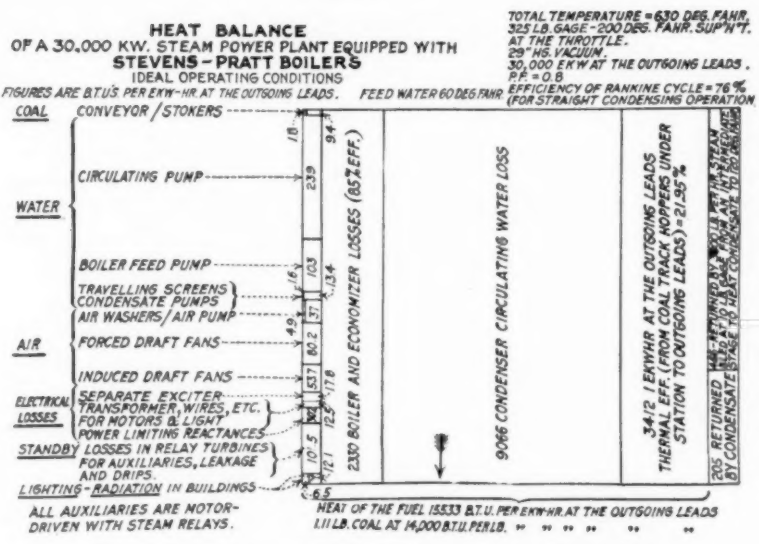


FIG. 4 HEAT BALANCE OF A 30,000-KW. STEAM POWER PLANT

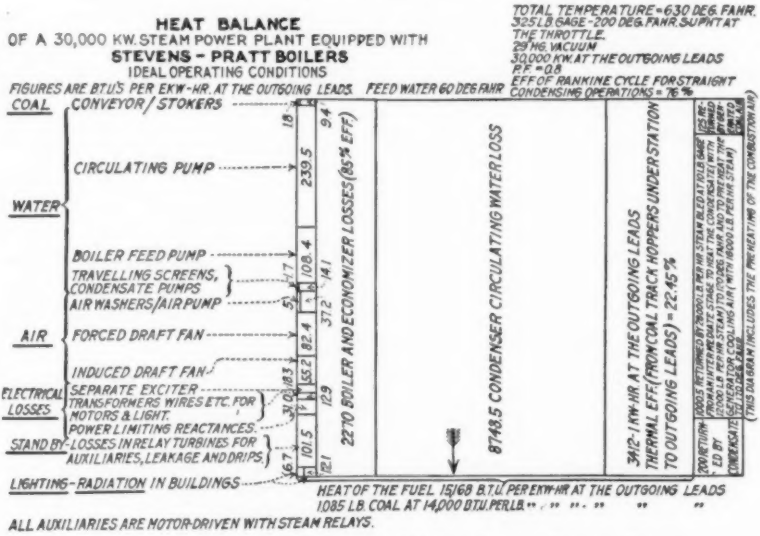


FIG. 5 HEAT BALANCE OF A 30,000-KW. STEAM POWER PLANT

652 deg. fahr. Under these conditions, the heat balance of a station properly designed and properly built would be about as shown in Figs. 4 and 5.

The plant should also be constructed for a boiler and economizer efficiency of 80 to 85 per cent by the use of every refinement known in the art where the stations have a load factor of 40 per cent or more.

In other words, the writer believes that American engineers should lead rather than trail the art, and if the English engineers are able to operate a station like the North Tees at 475 lb. pressure and a total temperature of 720 deg. fahr. and the French engineers, with a new 200,000-kw. station in Paris, are able to operate with 355 lb. pressure and 700 deg. fahr. total temperature, why should we lag in this country?

Further, in the judgment of the writer, the stations should be built entirely on the panel system and the separate panels tied in electrically only, so that as the art advances, more efficient stations may be built as the demand for power increases, to say nothing of the insurance possibilities of stations built on the panel system.

J. R. McDermet. It is interesting to contemplate the emphasis which is placed upon reliability by Mr. Hopping in his discussion of the Delaware station. In the writer's experience this has always been the final limiting factor in regulating the decision on the type of auxiliary drive. It is always possible to draw very fine conclusions on the adjustment of exhaust steam heat to boiler-feed temperatures, but invariably the calculations which look the most satisfactory from a purely heat economy standpoint, are deficient in reliability from an apparatus standpoint, and reliability still is perhaps the most desirable quantity. It appears to the writer that the system described for the Delaware station offers perhaps not the maximum economy, but it does offer both reliability and flexibility. Fortunately, also, the economizer installation, in which this station is unique among the others, offers more leeway and requires less flexibility than the heat-balance system offers.

It is possible to sub-divide systems of driving central-station auxiliary apparatus into three groupings:

1 All-steam
2 House turbines with electric distribution
3 Extraction heaters.

Each of these systems has had its vogue. The all-steam was originally the favored system for all large installations. At present it seems that the house turbine is very much in fashion and practically all

stations going into service lately have employed this method. However, the principal stations under contemplation are being designed with extraction heaters, taking steam from an intermediate stage of the main turbine.

From a purely theoretical standpoint the extraction heater offers the maximum efficiency. It is possible by rather complicated mathematics to derive the relationship that with an infinite number of extraction heaters following the turbine expansion, the power-station cycle reduces from the Rankine to the Carnot, which is the ideal cycle. This is due to rather a fortuitous circumstance, since the steam-table product of quality times latent heat is a constant for isentropic expansion, the maximum variation being less than 4 per cent of the mean value for Marks' and Davis' tables, and slightly less than that for Callendar's tables. Obviously, from practical considerations the infinite number of extraction heaters is impossible, and the calculated result is only of suggestive significance.

From a practical standpoint the extraction-heater method has certain manifest advantages, however.

The main turbine which has previously utilized the steam up to the extraction-stage pressure obviously has more ability to extract useful work than any smaller auxiliary. Recent figures on a 30,000-kw Curtis unit with extraction at the 6.5 absolute pressure stage gave a water rate of 13.6 lb. per kw-hr., which is considerably higher than any small turbine designed purely for carrying auxiliary load with electric drive.

Turbines for auxiliary purposes either of the small auxiliary or the house-turbine type with the modern ranges of superheat and pressures exhibit superheat of perhaps 100 deg. in their exhaust. This not only results in wasted energy but a very unfavorable condition is imposed upon surface heat transfer apparatus, such as evaporators, on account of the low conductivity of the superheated steam.

Both from the standpoint of low steam rate and desirability in connection with economizer work, the extraction point on the turbine is usually farther down in the staging than the saturation point of the steam. As a result the extraction connection may be used advantageously to remove undesirable water from the turbine and avoid erosion due to water in the later stages.

The pressure on the extraction connection varies considerably with load on the main unit, resulting in a variation of temperature of the water delivered, the temperature being of course lower at

low loads than at high. This is in a sense a partial automatic regulation of heat balance, and is particularly desirable in economizer installations where the economizer has the ability to care for higher heat ranges at low loads.

The obvious disadvantage of the employment of the extraction-heater method is the unreliability of the station against system disturbances when the station operates on the base load of the net work. All auxiliary power is derived from the main unit, and any disturbances in which the main unit participates immediately react on the auxiliaries. This objection in some cases is not of fundamental importance when the station is a part of a large net work, in which its total capacity is small in comparison with the aggregate. It is probably only to such conditions that the method is to be regarded as applicable.

It is the writer's impression that Mr. Hopping speaks with considerable insight when he remarks that boiler-room attendants and even turbine engineers probably do not understand the operation of electric equipment. It has been the writer's observation in the case of two stations which he has had an opportunity to observe intimately that the reliability of the electric drive has been low when the station was first put in operation. The explanation of this is that the apparent innocence of remote control apparatus has occasioned considerable meddling on the part of operators which has resulted in interruptions of service. Probably this meddling is to be considered in the light of lack of understanding of the apparatus rather than maliciousness.

The value which Mr. Hopping assigns to vent losses from the heater appears to the writer to be roughly correct, but if anything to err slightly on the side of being too low.

These figures are sometimes derivable from the air content of the feedwater entering the heater and can be checked from the standpoint of experience with such quantities. Heater vent loss is of course a necessary evil with high-temperature water where the purpose of the high temperature is to eliminate dissolved oxygen from the boiler feed and prevent corrosion. Necessarily, the solubility of oxygen in the boiler feed is a function of the partial pressure of the air in the heater space and has only an indirect relationship to the temperature of the water. Obviously, the only way to reduce the partial air pressure is to vent steam rather liberally to the atmosphere in order to secure a reduction in partial air pressure.

Mention is made by Mr. Hopping of a projected installation of

air extractors in this station. The installation of the air extractors from a heat-balance standpoint has three possible advantages:

1 The elimination of vent loss heat in the operation of the heaters. This value in the projected operation of the heater at 165 deg. fahr. in combination with the air extractor will reduce to approximately 0.5 per cent as opposed to the value of the 5 per cent given by Mr. Hopping

2 The loss of water through the heater vent in the form of steam will also be eliminated

3 The reduction of temperature of feed from 210 to 140 deg. fahr. will give a much added economizer effect and probably increase the boiler efficiency by 1.1 per cent beyond its present unusually high value of 80.4 per cent.

Fundamentally, of course, the idea of installing the air extractors is the prevention of economizer corrosion and the employment of lower feedwater temperatures to the economizer. It, however, has certain advantages other than the above in improving heat balance.

The air extractors which are to be employed operate upon the Elliott system of explosive boiling, in which water is injected suddenly into a region of vacuum and boiled at the expense of the heat of the liquid. The vapors and the heat liberated in the boiling process are recovered in a condenser which is cooled by the water before it goes to the heater. From an operation standpoint the heat consumption is determined by the temperature rise of the water as evinced by the temperatures entering from the supply and the water leaving the deaerator. The transfer of heat between the deaerator and the deaerator condenser, of which the heater secures the benefit in added water temperature delivered to it, however, may result in added temperature in the heater beyond the general design factor of 25 deg. which is assigned as the operating range for the deaerator. This in some stations might conceivably extend to the range covered by the extraction connection, depending upon the design of the component parts of the deaerator, and the deaerator will then automatically claim heat from the extraction condenser proportionate to the requirements of the heat balance, at the same time maintaining an unvarying temperature of economizer feedwater.

The flexibility and the close control possible under the auxiliary operation outlined in Mr. Hopping's paper have made this feature of

heat balance regulation of minor importance, however. The deaerators for the installation, therefore, are intended to deliver water to the economizer at 140 deg. fahr., with 165 deg. on the heater. The deaerator will, however, maintain a maximum temperature of 158 deg. fahr. on the economizer feedwater, when the pressure and temperature in the heater exceed the pressure of the extraction connection on the turbine. Therefore, the deaerator supplements Mr. Hopping's already careful system to the extent of permitting only 18 deg. possible temperature variation on feedwater fed to the economizer.

The writer ventures the opinion on feedwater temperatures in connection with heat balance that as steam pressures and temperatures continue to increase and boiler ratings are forced higher the subject of corrosion will reappear in boilers without economizers purely as a phenomenon of extremely high operating rating.

JOHN ANDERSON. The writer assumes that Fig. 3 of Mr. Hopping's paper is a study only and does not represent operating results. If it does represent the latter, why are economizer and boiler shut-downs, soot blowing, and miscellaneous plant losses included and banking coal omitted? It would be extremely interesting to compare actual operating figures, particularly those covering a month's operation, with the heat balance as calculated.

In the writer's opinion the boiler and economizer efficiency used by Mr. Hopping is higher than can be maintained with a feedwater temperature of 210 deg. fahr. entering the economizer. With such a temperature and operating the boilers at 176 per cent of their rating, as he indicates is the case, it seems as if the installation of an economizer is hardly justified.

Mr. Hopping has done some very close figuring to obtain the result of 17,982 B.t.u. per net switchboard kw-hr., as shown in Fig. 3. Using the same method of figuring and considering a station designed for the use of fuel in pulverized form, he would have no difficulty in getting a figure of 16,900, the stations being identical in design with the exception of the furnaces and other fuel-burning equipment.

NEVIN E. FUNK. In each of the three papers presenting figures for a heat balance, different methods have been used so that it is difficult to make comparisons. Should not the heat balance be standardized by the Society as boiler testing has been?

The figures for the Connors Creek plant of the Detroit Edison Co., and for the Delaware station of the Philadelphia Electric Co.

include the boiler and furnace, but those for the Colfax plant of the Duquesne Light System are from the boiler nozzle on, so that the B.t.u. per kw-hr., 14,200, etc., would probably amount to about 19,000 for the entire plant.

The impression should not be got from Par. 13 of Mr. Hopping's paper that the Philadelphia Electric Co. is averse to electric drive of auxiliaries. With automatic control apparatus so designed that push buttons to start and stop are the limit of complication in so far as the attendant is concerned, the desired results in control can be had with a minimum of attention from the operator.

The writer would like to take exception to the statements of Mr. Stevens regarding high pressure and temperatures. Minimum unit costs are not obtained by high investment costs and the resulting high efficiency alone, or by low investment costs with low efficiency and cheap fuel alone, but by a combination of investment and operating costs resulting in a minimum total. Trouble has been experienced in those stations using high pressures and temperatures, and the cost of shut-downs and repairs must be included in the cost of current before the theoretical saving can be justified. The writer believes that there is a general trend toward higher pressures and temperatures and that progress along this line has been made so rapidly that the experience gained has yet to be weighed. While figures on the costs of high-pressure and high-temperature plants that now seem to be European practice are frequently quoted, the operating figures for these plants do not seem to be readily available, but without them a practical justification of these plants is not at hand.

Leo Loeb. The writer is impressed with the results which the Philadelphia Electric Co. and the United Electric Light & Power Co. have obtained in arriving, directly and simply, at heat balances for maximum economy.

Mr. Hopping has wisely emphasized the important consideration in selecting auxiliaries when he states that it would be unwise to take advantage of the economies made possible by the use of electric drive of certain auxiliaries unless reliability of the supply of energy for the drive and the sturdiness of the equipment were comparable to that of steam drive. In addition to this are two secondary considerations:

a Whether the cost of the more complex systems is warranted when translating the fixed charges on increased

investment and the operation and maintenance to switchboard cost per kilowatt-hour under the normal load factor of the station

b A full consideration of the inherent heating losses in electric generation, distribution and motor utilization, which are non-recoverable, when the source of energy for electric drive is a unit other than the main turbogenerator, as against the reduction in effective peak capacity of the units when drawing on them for auxiliary power.

The economy of the prime mover is determined by pressure, superheat, vacuum, size, and factors of cost which the power company is willing to incur to obtain a desired water rate. The remainder of the heat-balance problem of the central-station designer is to obtain that effective combination of steam auxiliaries which will have the lowest water rate to yield the steam necessary for feedwater heating. In order to develop this point, Table 2 is presented.

TABLE 2 SUMMARY OF HEAT BALANCE, DELAWARE STATION

Average load on station, kw.	44000 *	30362 †
	Per cent ‡	Per cent
Boiler losses	19.0	20.3
Plant steam losses	3.4	3.7
Banking loss	——	0.4
Circulating water loss	56.4	56.0
Steam auxiliaries — energy	0.6	0.6
Electric auxiliaries	0.6	0.8
Generator and bearing losses	1.0	1.2
Net generator output	19.0	17.0
Heat supplied by coal	100.0	100.0
Returned to boilers from waterbacks, condensate and steam auxiliaries	12.9	12.3
Turbine water rate — per kw-hr. generated, lb.	11.1	11.7
	Per cent	Per cent
Electric auxiliaries, including excitation	3.2	4.6
Steam auxiliaries, energy	3.1	3.7
Total auxiliary load	6.3	8.3

* Most economical load.
† Actual figures for one week's run during April 1921.
‡ Corrected for heat reclaimed in waterback.

Mr. Lawrence has clearly indicated the importance of low water rates for station auxiliaries as one criterion of an effective heat balance in a central station. Fortunately, the attitude formerly held

by some operators that the water rate of auxiliaries was of slight moment since the heat was recoverable in feedwater has almost disappeared, and with it there has developed an incentive for the small-turbine builder to produce an efficient as well as reliable unit.

The Delaware station data show that over one week's operation the hourly auxiliary steam consumption of 46,445 lb. produced a feed temperature of 204 deg. fahr. when converting 3,762,045 B.t.u. at a water rate of 31.4 lb. per brake hp. into useful work of 1478 hp. The electrical equipment of this energy figured back to generator output is about 1306 kw. or substantially equal to the energy actually used by motor-driven auxiliaries, the latter energy requiring 15,650 lb. of steam per hour for its production. Any improvement in auxiliary water rate has the affect in a duplex drive of transferring load to steam auxiliaries, reducing the steam required for the main unit, and increasing the generator capacity for net station output. Thus the duplex drive should be more efficient than the house turbine (eliminating any advantage of size factor as affecting water rate of house turbine against individual auxiliary turbine) since the power generated by the auxiliary turbine is applied directly to the working shaft, avoiding the inherent electrical losses of excitation, generation, distribution and motor utilization, which may easily amount to 25 per cent.

In order that the results from Colfax and Hell Gate may be compared with those from Delaware and Connors Creek it is highly desirable that they be submitted for a closed period of either one week or one month.

FRANCIS HODGKINSON. The writer thinks he is correct in saying that the earlier plants in this country operated their auxiliaries by means of electric motors receiving energy from the station system. Because of the unreliability of both the motor and the source of power, there were many interruptions. Plainly, when there is line trouble or anything of that kind, that is the very time when the condenser auxiliaries and such things should be most dependable. This naturally brought about the adoption of steam drive for auxiliaries. The exhaust of the steam auxiliaries was naturally employed for heating feedwater, and engineers very generally deluded themselves that inasmuch as the exhaust steam was condensed in a feedwater heater, the thermal cycle was very high and the consumption of steam per horsepower of the steam engines driving the auxiliaries was immaterial. Steps were taken to proportion the num-

ber of non-condensing auxiliaries in the power plant to give the minimum waste of heat, and it was customary to so proportion these things that the feedwater was heated to about 212 deg. fahr., with three-quarter load on the main unit. This was probably the best approximation, but it meant waste of steam to the atmosphere with a lesser load on the main unit and colder feed at higher loads at a time when hot feed was more important.

About ten years ago, the writer devised a series of valves for accomplishing a heat balance. These were built in a variety of ways, some of them merely providing for admitting the surplus exhaust steam, beyond that taken by the feedwater heater, into a low-pressure stage of the turbine. Others provided for both bleeding the turbine at a time of deficiency and taking steam into the turbine when there was a surplus of steam for feed heating. These valves would be quite satisfactory today for use in moderate-sized plants.

Engineers are now generally operating their auxiliaries by means of electric motors. One of the principal reasons for this is the avoiding of congestion and the absence of the beastly mess that so frequently obtains in the basements of power houses on account of the enormous amount of piping with drips, drains, etc., which, together with the high temperature of the basement, usually results in neglect of the apparatus. Unquestionably, today, the electric motor is just as reliable as any steam drive.

Mr. Hopping's paper describes circulating pumps driven by both an electric motor and a turbine. This arrangement, of course, permits the maintaining of a heat balance, but it is not free from the objection of the steam drive cited above.

It is very evident that the greatest economy is to be secured by bleeding the main turbine in stages, the number of stages depending upon the degree to which it is desired to heat the feedwater, this latter depending upon whether the plant operation has warranted the installation of economizers. The motors operating the auxiliaries would then receive their energy from the main power house system. While this is plainly the most economical thing to do, it carries a serious risk to reliability, for should there be line trouble, it would interfere with the operation of the auxiliaries at a time when they are most needed, all of which would seem to lead one to favor the house-turbine scheme.

There is a feature of the Cheswick Power installation which Mr. Clarke describes that particularly appeals. The writer refers

to the motor-generator set for preserving heat balance, and a feedwater temperature thermometer located on the switchboard so that the switchboard operator regulates the feedwater temperatures by adjustment of the energy transferred through this motor-generator set, which has been called a "heat-balance set."

There is another point closely allied to the general subject of heat balance, although not exactly relevant, namely the elimination of oxygen from the system, which is an important factor in the prevention of corrosion. A feedwater heater and a condenser are synonymous. The steam from the house turbine is condensed in a piece of apparatus which may equally well be termed a feedwater heater or a condenser. It would seem to be important that this is provided with an air pump, so that water leaving this condenser, or feedwater heater, will be in a state of evaporation, so that it is entirely relieved of oxygen before its passage to the feed pumps, economizers, boilers, etc.

The writer understands there are some installations where corrosion has been a serious trouble and where it has been entirely stopped by passing the hot feedwater through a vessel in which there are iron chips or screens which absorb the oxygen by becoming corroded.

H. R. SUMMERHAYES[1] said that for a long time there had been a delusion that steam-driven auxiliaries were the most efficient, but gradually the higher maintenance of steam-driven auxiliaries made electrical operation necessary. He also called attention to the necessity of designing steam-driven units for the lowest steam consumption in order to secure maximum economy in the plant.

R. J. S. PIGOTT. The papers presented by Messrs. Hopping, Clarke and Lawrence consider the heat balance of a plant chiefly at a certain point of load. In deciding upon a type of drive for auxiliaries, the heat balance at a single point of load may be, in some respects, misleading, on account of the fact that at various loads on the station the heat balance is very differently distributed. It is possible with a certain type of auxiliary drive to obtain the lowest heat consumption at best load, but the average heat consumption occurring with a given load curve may be higher than with some other combination. The writer has no doubt that these gentlemen have worked out the total heat input-load output lines over the whole range of load to be carried by the station, and it would have

[1] Engr., Lighting Dept., General Electric Co., Schenectady, N. Y.

been extremely interesting to see the comparison of these various forms of auxiliary drive schemes, not only in the total heat input-output curves, but also in the B.t.u. per net kilowatt-hour curves over the complete load from zero output to full capacity of the stations as designed.

As an illustration of the point, similar curves drawn in the writer's paper [1] on Graphic Methods of Analysis in Design and Operation of Steam Power Plants, Figs. 13 and 15 illustrate the difference between all-steam and all-electric auxiliaries, as arranged for this particular installation. The shape of the curves is quite as important as the best load values.

Another point of interest to everyone, which has evidently occupied the minds of these gentlemen in laying out these systems, is the question of reliability. No method of definitely evaluating reliability has yet been devised, but it is certain that the increase in reliability of electrical apparatus, especially controls, has reintroduced the electric drive to a very large extent, whereas but a few years ago electric drive could certainly not have been considered satisfactory, in spite of its obvious advantages. These papers give us a crystallization of present opinion from the best qualified sources, but they do not give completely the reasoning that caused the adoption of any of the systems in a given case.

L. P. Breckenridge said that he was impressed with the importance and the desirability of serving great industrial areas or so-called zones by power plants in which the heat energy of our coal shall be converted into electric energy in the most economical way, and called attention to the Superpower Survey report. [2]

T. E. Keating. For each set of steam operating conditions and auxiliary system used there should be some definite temperature of boiler feed at which the overall station efficiency is the maximum. Fig. 8 in Mr. Berry's paper indicates for his station a value of approximately 160 degrees without the use of economizers. It is the writer's opinion that this value is better suited for operation with about 50 per cent economizer surface.

Mr. Hopping states in one part of his paper that at the present time he is operating with 210-degree feedwater, and later states that the overall plant efficiency will be improved by operation at 140 degrees in connection with economizers and air extractors.

[1] Trans., v. 38, 1916, p. 687.

[2] Professional Paper No. 123, U.S. Geological Survey.

A study of the method of arriving at the most desirable boiler feed temperature for maximum station efficiency both with and without economizer surface, together with the comparative investments for the respective installations, would be of great value in this heat-balance problem.

JOSEPH POPE said that his experience with the duplex drive had been that it presented difficulties in dividing load between the drives in any proportion other than all on one or all on the other driving unit. He also spoke of the lack of speed control on large turbines at the auxiliary steam inlet connection which might result in overspeeding in case of a very light load on the turbines and an excessive supply of low-pressure steam.

JAMES B. SCOTT asked if the authors of the papers would install the systems of heat balance described by them in case the stations were to be redesigned.

GEO. A. ORROK. These papers are extremely interesting and the diversity which they show is even more interesting than the papers themselves. The question of heat balance, however, is not really fundamental. The fundamental question is the use of heat. Power stations are heat engines, they are amenable to thermodynamic laws and the Carnot cycle represents the maximum economy which can be obtained in a power station where heat is transformed into work. Unfortunately the steam engine or turbine is not adapted to the use of the Carnot cycle, but the Carnot cycle can be approached by taking heat from the expansion side of the cycle on the entropy diagram and delivering it to the intake or heating portion of the cycle. There have been many devices suggested for approximating this condition. Feedwater heaters, economizers, the open and closed heater systems with bleeders from various portions of the turbine flow are examples of this sort and, theoretically, the problem would be solved by an infinite number of feedwater heaters.

Regarding the question of auxiliaries, we have seen three types of station design come into use with remarkably good results from each type. Steam-driven auxiliaries, the earliest form, had much to recommend them, but the electric auxiliaries and the type of station with a house turbine which has been termed the "heat trap" variety are also very attractive. Load factor is really the determining factor in the selection of auxiliaries and the writer is not yet clear in his mind but that for certain load factors steam-driven

auxiliaries offer the best chance for satisfactory design and operation. Thus the field of power-station design might be plotted in an equilateral triangle in which one side represents the base for all-steam auxiliaries, a second side all-electric and the heat-trap arrangement for the third. Somewhere within this triangle all arrangements of auxiliary apparatus would be included and the point of maximum economy is still in doubt.

JAMES M. TAGGART. In the papers treating of heat balance anyone familiar with power-station practice will recognize an advance. In such figures as submitted one can find no proof of what has been attained. The subject of heat balance as it is designated is of necessity tied up with the heat return of the economizer and the efficiency characteristics of the steam-driven units from which the exhaust steam is taken to heat the feedwater.

Mr. Hopping in his paper states that a better efficiency is expected at the Delaware station when the temperature of feedwater from the heater is dropped to 140 deg. fahr. Evidently an increased heat return from the economizers is expected as a result of this reduction in temperature of the feedwater. It would be of interest to know how much increase in heat return is realized, also what the increased transference in the economizer per square foot would be. The question of pitting and corrosion of feed piping and economizer tubes is a chemical one. In many cases this is found to be due to a slight acidity in the water rather than to the presence of contained air.

Mr. Berry, in the writer's opinion, is on the right track when he emphasizes the importance of speed control of auxiliaries. At the present time practically all power-station auxiliaries are rotary. In the case of the draft fans a wide variation of speed is absolutely needed to give an economic operation. In most power stations also all the condenser pumps should have some speed control to obtain the most economic operation and best to meet the varying load conditions. Whether it will be possible in the future to obtain as suitable regulation with the alternating-current motors as can now be obtained with direct-current motors is questionable. The curve which Mr. Berry shows indicating the most economic temperature for feedwater is of interest. The writer has figured out a similar curve which was published in the August 1919 number of *Steam* for a condition of 300 lb. initial pressure and approximately 0.35 lb. back pressure. The exhaust steam for heating the feedwater was

figured as taken from the main unit, namely a high-pressure engine and low-pressure turbine. In this case the theoretical maximum temperature of the feedwater without economizers and with a single heater was found to be approximately 250 deg. fahr. For every variance in initial pressure, vacuum, superheat and type of steam drive this temperature of maximum economy will vary. When economizers are used, various new factors are introduced such as the size, efficiency of heat transference, temperature of flue gases and added power required for induced draft.

With an efficient engine drive for the house generator the temperature of feedwater for maximum efficiency will be much higher than with the turbine. This follows from the fact that for the capacities considered an engine gives a higher power return at high-pressure ranges. For smaller auxiliary power the difference would increase.

Mr. Clarke's description of the heat-balance system at the Colfax station would indicate the fact that excellent control of the feedwater temperature at all loads was being obtained. In using the turbine for the forced draft and boiler-feed pumps considerable economy is being sacrificed in order to secure a problematical increased reliability. In an extended experience the writer has found that turbine drive for these auxiliaries is generally run very much under-loaded. This condition is one of very low steam economy for the turbine drive used. Mr. Clarke's curves of boiler efficiency while perhaps aside from the subject are very interesting. It is noteworthy that at the lower loads the efficiency and temperature of the flue gases both fell off rapidly. This would indicate evidently an increase in excess air either due to poor air regulation or to excessive leakage in the boiler setting.

In all the systems described the attempt to secure an increased reliability of service has led to some complexity. It is to be hoped this may be simplified in future systems.

M. D. CHURCH. In deciding between motor-driven and turbine-driven auxiliaries we must not lose sight of the fact that the energy losses in the generator and motor are seldom less than 20 per cent of the average power developed by the motor and that this loss can not be recovered, as can the heat in the exhaust steam from a turbine.

The steam consumption of turbine-driven auxiliaries has been practically cut in half by the use of reduction gears as compared

with direct-connected units. The higher speed of the turbines gives a better steam consumption per horsepower-hour, and the driven machine may be run at the speed which gives the highest efficiency. This latter is possibly the more important as the energy lost due to inefficiency of a blower or circulating pump is not recoverable, whereas the exhaust from the turbine still contains practically all the heat which was in the entering steam except that actually converted into useful work, as shown in Mr. Berry's paper.

Reduction gears are now available which are fully as reliable as any other piece of power-plant apparatus. This has been brought about by more logical ratings, the use of properly heat-treated gear forgings, accurate construction and more positive lubrication.

Mr. Berry shows in Par. 12 that whereas a very efficient large condensing turbo-generator uses from 18,000 to 21,000 B.t.u. per kw-hr., a house turbine, because the heat in its exhaust is utilized, produces a kilowatt-hour for about 4550 B.t.u. This is a very concrete and quantitative expression of the general statement frequently made that when the heat in the exhaust steam can be fully used, a non-condensing turbine can produce power at a fuel cost far below that of any condensing turbine, however efficient.

Mr. Hopping, in Par. 7, brings out the large degree of heat-balance adjustment as well as the increased reliability which may be obtained by the use of duplex drive on the larger auxiliaries.

Some question has been raised concerning wear of the governor valve when the load is divided in adjustable proportion between the turbine and motor on duplex drives. As a member of the Terry Steam Turbine Company, which has supplied by far the greatest part of the duplex drives installed to date, the writer has found that there is no more trouble encountered due to valve cutting than on any turbine which operates a large portion of the time at low or partial loads. Governor valves for duplex drives are now being made of materials particularly suited to withstand wire-drawing effect.

LINN HELANDER. Proper feedwater temperatures for power plants using feedwater heaters of the condenser type may be readily determined by considering the total station load as including that of the steam- as well as electrically-driven auxiliaries. By assuming feedwater temperatures, this load then may be divided variously between that developed by steam, the latent heat of which subsequently is dissipated, and that developed by steam, the heat of

which is returned to the feedwater heater. Computations to determine, for each feedwater temperature, the total heat consumed in developing the station load, then, will indicate the feedwater temperature at which the fuel charges are minimum.

For the case analyzed by Messrs. Berry and Moreton, the station load is divided between the main unit and the house turbine. With this arrangement, increasing the load carried by the house turbine equally reduces the load carried by the main unit and consequently reduces the steam consumed by it. The reduction in steam will be equivalent to the reduction in load, multiplied by the slope of the main units' Willan's line. Calling the slope of this line N and the reduction in load dL, the reduction in steam is NdL, and, if H is the heat content of the steam above that of the condensate, the reduction in heat consumed by the main unit will be $HNdL$. Representing the heat required by the house turbine to produce a kilowatt-hour as h, the reduction in the heat consumption of the station effected by transferring load dL to the house turbine is $\frac{HNdL - hdL}{e}$, e being the boiler-room efficiency. So long as the heat required by the main unit to produce a kilowatt-hour is greater than that required by the house turbine, i.e., so long as HN is greater than h, the heat consumption of the station will be reduced by increasing the load on the house turbine and decreasing it on the main unit. Best economy will be realized, then, when the load on the house turbine is maximum. Neglecting radiation and friction losses, as was done by Messrs. Berry and Moreton, the heat required by the house turbine to produce a kilowatt-hour is 3415 B.t.u. The minimum value given in their Table 3 for the slope, N, of the Willan's line is 7 lb. per kw-hr., and, since the heat content of each pound of steam measured above that of the condensate is 1275 B.t.u., the minimum heat required by the main unit to produce a kilowatt-hour is 7 times 1275 or 8925 B.t.u. This is 5510 B.t.u. greater than that required by the house turbine. Best economy should, therefore, be realized when the load on the house turbine is maximum.

The feedwater temperature at which the heat consumption of the station is minimum may be determined, as Messrs. Berry and Moreton have done, by determining the load carried by the house turbine at various feedwater temperatures and computing the corresponding station fuel charges. In their Table 3, however, Messrs. Berry and Moreton indicate that the feedwater temperature

for best economy is not that at which the power developed by the house turbine is maximum. In the body of their paper, they justify this conclusion by the inference that influences bringing this about creep in as the exhaust pressure increases. If these influences include radiation losses, leakages, costs, etc., that may increase as the back pressures increase, the statement has justification; but, on the basis that these items are neglected and that the only influence opposing the improvement in station economy is that of increasing water rates at high pressures, it is not correct. This increasing water rate largely does determine the temperature for best economy, but this temperature, nevertheless, is that at which the house-turbine load is maximum. Further study of Table 3 indicates the probability of numerical error in the item for the cost of power with a feedwater temperature of 160 degrees. Also, plotting the loads carried by the house turbine against the feedwater temperatures indicates that the proper feedwater temperature is about 190 deg. fahr.

Messrs. Berry and Moreton use the formula

$$H_x = H_i - \frac{3415L}{S} - R$$

for determining the heat content of the exhaust steam from the house turbine. The load L in this formula should be the indicated load and should include the mechanical losses of the turbine and generator.

Using economizers, the problem is not much different, except that the reducing efficiency of the boiler and economizer must be weighed against the increasing efficiency of converting steam to power as the feedwater temperatures increase. This will reduce the feedwater temperature for best economy, but, inasmuch as changes in economy resulting from variations in the temperatures of the feedwater are small in the neighborhood of best economy, it is likely that reducing this temperature so as materially to enhance mechanical difficulties will not be warranted. With steam pressures and superheats usually advocated for central stations, the temperature of the feedwater for best economy probably will be considerably above that at which sweating occurs.

E. L. Broome. In connection with the three able papers upon the subject of heat balance in steam power plants, it may be of interest to present a few operating statistics from the Glen Lyn

Steam Plant of the Appalachian Power Company, in which plant the main auxiliaries are operated by steam turbines.

It is not the writer's purpose to advocate steam-driven auxiliaries, but rather to show for the purpose of comparison what may be expected from such an arrangement in a plant of moderate capacity under operating conditions which were unusually favorable.

TABLE 3 OPERATING STATISTICS, GLEN LYN STEAM PLANT, APPALACHIAN POWER CO.

	September	October	November
Hours in month	720.0	744.0	720.0
Hours in operation	676.0	692.0	611.0
Average pressures and temperatures			
Gage pressure at boiler, lb.	270.8	267.8	267.4
Gage pressure at turbine, lb.	252.9	251.7	251.3
Barometer, in.	28.6	28.6	28.6
Absolute back pressure, in.	1.6	1.1	0.9
Superheat at turbine, deg. fahr.	162.8	167.4	174.8
Hotwell temperature, deg. fahr	93.2	77.9	72.0
Feedwater temp. at heater, deg. fahr.	200.9	192.4	184.4
Gas temperature, main flue, deg. fahr.	476.0	455.3	454.8
Calorific value of coal as fired, B.t.u. per lb.	13,150.0	13,771.0	13,625.0
General results			
Evaporation per lb. coal as fired, lb.	8.7	8.57	8.44
Evaporation per lb. coal from and at 212, lb.	10.2	10.16	10.07
Boiler rating, per cent	172.0	181.0	180.0
Boiler and furnace efficiency, per cent	75.5	71.6	71.7
Turbine economy, lb.	11.3	10.9	11.0
Station economy, lb.	13.2	12.6	12.70
Coal per kw-hr., lb.	1.51	1.47	1.51
Load factors — 720 hours			
Based on turbine rating, per cent	67.1	67.8	63.4
Based on maximum hourly peak, per cent	74.7	71.3	63.4
Load factors — 676 hours			
Based on turbine rating, per cent	71.8	73.6	74.8
Based on maximum hourly peak, per cent	80.0	77.6	74.8
Kw-hr generated	9,672,000.0	10,190,000 0	9,142,000.0

The plant at present consists of one 15,000-kw. and one 20,000-kw. General Electric turbine and four cross-drum B. & W. boilers of 12,320 sq. ft. of heating surface each. There are three more boilers of the same capacity under contract. The stokers are Westinghouse underfeed, fourteen retorts wide.

The condenser auxiliaries, the boiler feed pumps and the stokers and fans are all driven by steam turbines. The coal- and ash-hand-

ling equipment, miscellaneous pumps, etc., are motor driven; the exciters are direct connected.

The steam plant was originally designed as a relay to two water-power plants, and as is usual in such cases, low investment cost and simplicity, combined with reliability, were prime considerations. The unusually protracted low-water period and a rather favorable increase in connected load during the three months covered by the statistics resulted in load factors which are admittedly considerably above the anticipated average.

JOHN WILSON BROWN, JR. In discussing the performance of the house set, the authors of the paper on the Connors Creek plant say: "Clearly the displacement of a portion of the power generated by the main units at high cost by power generated by the house turbine at much lower cost will result in a gain in station economy. From this it would appear that the maximum house-turbine output is to be striven for. But the matter is not so simple." Now the fact is that if the wording is changed a trifle to read "the maximum house set output provided no exhaust from the house set goes to waste is to be striven for," the matter is exactly "so simple" and it requires no complex calculation to prove it.

If L = total load

L' = load on house set costing a B.t.u. per kw-hr.

then

$L - L'$ = load on main units costing b B.t.u. per kw-hr.

Total cost of load will be $b(L - L') + aL'$

Cost per kw-hr. will be $b - (b - a)\dfrac{L'}{L}$

It is evident, if $(b - a)$ is positive, that this value decreases continuously as $\dfrac{L'}{L}$ increases. The most economical point then is when $\dfrac{L'}{L}$ is a maximum. For an existing plant the quickest and most reliable way to determine this point is to vary the back pressure on the house set and compare the readings of the total-load ammeter and the house-set ammeter. This is much surer and less laborious than any calculations.

If the result determined by either method shows 160 deg. fahr.

to be the most economical temperature for the feedwater, it constitutes a condemnation of the judgment of the man who designed the plant.

The writer does not doubt that, with the turbine set assumed by the authors, the limiting value of the energy to be obtained from the house set would be about as the authors have stated it. But in such an equipment as they have outlined there still remains the opportunity by installing additional apparatus to generate at 4550 B.t.u. per kw-hr., 2000 kw. by heating the water from 160 deg. to 210 deg. fahr. and about 2500 kw. more by heating from 210 deg. to 300 deg. fahr.

The method of computation in Appendix 2 is unnecessarily complicated. After the first five steps there are known

L the total load
L' the maximum load on the house set
b the cost in B.t.u. per kw-hr. of power developed at load $(L - L')$ by the main engines.
Cost of power developed by the house set assumed at 4550 B.t.u. per kw-hr.

Then

$$\frac{4550L' + b(L - L')}{L} = \text{cost of total power in B.t.u. per kw-hr.}$$

J. KENNAL.[1] The subject of the use of high pressures and temperatures of steam in large power plants is being very much discussed at the present time, some engineers contending that the increase in pressure does not compensate for the additional cost of the plant, and some arguing that it is only the superheat temperature that counts, but one of the points respecting which we seem to have arrived at finally in the meantime is the temperature, for it is found that above a total temperature of the steam of 700 deg. fahr. the durability of the superheater is not all that could be desired, and that it is best in this respect to keep to a temperature of 650 deg. fahr., — the difficulty being the material.

There is no doubt that, by using an alloy with a large percentage of nickel for the tubes, higher temperatures can be worked with, but that is not within practical range yet.

We have had some experience with a pressure of 275 lb. and a

[1] Managing Director, Babcock & Wilcox Limited, London, Eng.

total temperature of 680 deg. fahr. at the superheater outlet, and with these limits an improvement in results has been obtained.

We have used higher pressures and temperatures in some instances, but the plants have hardly been working a sufficient time to allow of definite figures being obtained. We can say, however, that with a pressure of 350 lb. any trouble with the steam generating plant and the piping can be avoided.

We have working at the present time part of an installation of ten boilers, to give a total evaporation of 350,000 lb. of steam per hour. Some of these boilers have now been at work for several months, the remainder are in course of erection. The working pressure at the boiler is 350 lb., and the temperature of the steam leaving the superheater is 750 deg. fahr., the pressure at the turbine being 340 lb. and the temperature 725 deg. fahr.

We have also supplied a plant of ten boilers to give a total evaporation of 420,000 lb. of steam per hour at a pressure of 475 lb., and a temperature of 700 deg. fahr. This plant has now been at work for some months, but we are not at liberty to give any particulars concerning it, beyond stating that there has been no difficulty at all as regards the operation at the high pressure, neither has any difficulty arisen in regard to the keeping of the joints of the piping tight.

We have also in course of erection a plant of ten boilers to give a total evaporation of 1,000,000 lb. of steam per hour, at a pressure of 375 lb. and a final temperature of 740 deg. fahr.

Another large plant is being installed in Paris, where the working pressure is 355 lb. per sq. in., and the temperature of the steam leaving the superheater is 375 to 400 deg. cent. (780–830 deg. fahr.), but this has not yet been put to work, so it is impossible to give any information concerning it.

W. McClelland [1] wrote as follows: "It is evident that the use of pressures above 300 lb. per sq. in. and temperatures above 600 deg. fahr. is at present quite rare commercially.

"I have ascertained that the following plants are in operation, and, except where otherwise stated, as far as I know no serious trouble has arisen in connection with their running.

[1] Director of Electrical Engineering, Admiralty, Whitehall, London, S.W.I.

Situation of plant	Steam pressure of boiler, lb. per sq. in.	Temperature at turbine, deg. fahr.	Superheat at turbine, deg. fahr.
Birmingham	325	700	280
Manchester	375	700	260
Wales	375	750	315
Rugby (B. T. H.)	375	700	265
Durham	450	700	—
South Shields [1]	450	750	—
Paris	330	660	235

[1] Considerable trouble experienced, especially with steam joints.

"The usual working pressures throughout the country are between 200 and 300 lb. per sq. in., and the steam temperatures do not usually exceed 500 to 550 deg. fahr.

"It is probable that in the near future in the large stations now contemplated, the pressures of boilers will be considerably above the present average."

E. L. Hopping. In answer to a question which has been asked about burning fuel oil, it may be said that in laying out the Delaware station, three types of fuel were considered, coal, pulverized coal and fuel oil. Conditions at Philadelphia made the choice of stoker-fired coal most economical, but the furnaces were designed so that the volume would be sufficient for either pulverized coal or fuel oil and so that either of these methods might be employed with a minimum of change.

In answer to Mr. Pigott's criticism that the heat balance should be determined for more than a single load, it should be stated that an analysis of this question at varying loads was undertaken during the design of the station.

The feedwater temperature of 210 deg., mentioned by Mr. Keating, is that obtained under the present conditions of operation, and is necessary from a physical standpoint, due to corrosion in the tubes. The boilers in this plant have a 25 per cent economizer surface included in the boiler setting. With this type economizer the cost of installation, and therefore the original cost of the economizer, is a comparatively small item.

In answer to Mr. Scott's question about duplicating the present installation, it should be remembered that in any new plant the accumulated experience of all previously designed plants will shape the choice of equipment.

Referring to Mr. Stevens' discussion and his reference to the use of high pressures and temperatures abroad, the writer would like to state that in conversation with certain English engineers he had obtained the impression that while high pressures and temperatures had been used in some plants, the equipment, such as piping, valves and other parts coming in contact with the pressures and temperatures, was of a special nature, and it should be borne in mind that unless there is standard commercial equipment available for successful use in connection with high pressures and temperatures, the economies which may be obtained by such practice would be more than offset in the final analysis by the first cost of the equipment.

Mr. Taggart has referred to the increased efficiencies which may be expected with the use of feedwater at a temperature of 140 deg. fahr., as balanced against the present practice, in the Delaware station, of feeding the boilers with water at 210 deg. fahr. The increase in overall efficiency has been calculated to be about 1.2 per cent on the station coal rate. Practically all of the savings in this figure are obtained at the economizer and are due to the lower exit gas temperatures.

In reply to Mr. Anderson's question as to the boiler efficiencies, attention is called to the fact that the heat distribution shown on Fig. 3 illustrates not a theoretical condition but indicates what results could be obtained with the plant, as it is operating at present, if the load conditions at the station were more favorable. The banking hours shown on Fig. 4 are only necessary because of the extreme low points to which the load dropped for considerable periods throughout the week illustrated

Referring to the discussion by Mr. McDermet, as to the operation of motor-controlled apparatus in the boiler room, attention is called to the fact that this condition was foreseen at the time the plants using this type of control were designed and the apparatus was, therefore, protected in such a manner that it could not be affected by ignorance of operation of electrical apparatus by any of the operators in the boiler room.

Mr. Hodgkinson refers to the disagreeable conditions which obtain in the auxiliary basement where steam-driven auxiliaries are used. In the opinion of the author, this condition is largely one of housekeeping and partly of correct methods in installation, such as providing drip pans at the base of the machines and trapping some of the drips. While a steam installation is not quite as clean

as where motors are used, it can be kept in such a condition that it will be far from disagreeable.

C. W. E. CLARKE. "I would like to say in reference to Mr. Stevens' comment that in building stations like Colfax, Hell Gate, Delaware, Waterside and Connors Creek, we are not building engineering monuments. We are building stations to turn out current at the lowest cost per kilowatt-hour at the switchboard, and until coal prices go up much higher than they are now, steam pressures will stay about where they are. It is interesting to note that the Commonwealth Edison Company of Chicago, the controlling interests of which have a high-pressure and high-temperature station at Joliet, is building a new station at Calumet for a considerably lower total pressure and temperature.

"Mr. McDermet questioned the corrosion increase in boilers with rating. As a matter of fact, the activity of the dissolved oxygen in a boiler is very low when steaming. Most of the damage is done during banking hours.

"Mr. Funk asked about the boiler efficiency curves. The boiler efficiency for three months, that is September, October and November, averaged 78 per cent.

"The statement of coal per kw. was kept out of the paper purposely, and I expect at a later date to cover that question. Under present conditions, the station is operated on a gross output basis of 17,600 B.t.u. and on a net basis of 18,500 B.t.u. per kw-hr.

"Mr. Pigott called attention to the study of the heat-balance problem with a varying load. I take it for granted that he means a varying load factor. The reason for the selection of the system at Colfax was the fact that it appeared to be the most flexible, the most reliable, and covered a greater range of variation than anything we could put our fingers on.

"Mr. Pope asked about the total cost of the heat-balance system. That is rather a hard question to answer. The total cost per station should be the criterion. It is not fair to pick out one item in the station and say this costs too much. You must analyze the whole station. If the whole station is producing at a reasonable cost and produces current at the switchboard at a reasonable cost, that question disappears. The speed change in the system is three cycles. The only machines on that system that would be affected at all are the exciter units. Under proper regulation no difficulty is encountered.

"Mr. Jurgen, of Waterbury, questioned the duplex drive. I am not concerned with that especially, but if governor valves are kept ground and just a little attention is paid to the unit, there will be no trouble in keeping it operating as it should.

"In regard to Mr. Scott's question as to future units, I can only say that at Colfax the second system is being installed, and it is to be a duplicate of Unit No. 1 except for minor improvements.

"Mr. Orrok states that he thinks that the steam drive can be built with more economy than the electric drive, but I do not believe it can be done. The inherent losses, not only of heat and water, but other similar losses, I think will defeat his object."

C. H. Berry. Mr. Stevens' assertion that American engineers should lead rather than trail the art of power-station design and operation will meet with approval on the part of all who chance to be Americans. From this, however, one must not conclude that American practice should forthwith embrace maximum steam pressures and temperatures. The best engineering is not the most daring or the most advanced — it is rather the most suitable, and, as Mr. Funk has observed, the lowest production cost is attained only in a balanced design. Fuel costs less here than in Europe, so that the best point for us may quite possibly correspond to a somewhat lower thermal efficiency than is called for in Europe. Safety and realibility are of paramount importance, and to sacrifice either or both of them for a comparatively small gain in thermal economy would be the height of folly.

Mr. Funk seems to feel it necessary to reduce the control of electrical apparatus to the pushing of buttons if the control is to be effected by the plant-operating men. The experience of the Detroit Edison Company does not bear this out. We do not favor highly automatic devices, because of their decreased reliability; we therefore employ manually operated controls, educating our men to handle them properly. We have found no difficulty in doing this, and the results are altogether satisfactory.

In reply to Mr. Pigott, the table shown for the Connors Creek plant presents average results for one month of operation, on the usual commercial load. It is true that the computation for determining the best boiler feed temperature and auxiliary exhaust pressure is for only one set of conditions. Mr. Pigott very properly points out that this should be computed for various conditions, and the results averaged. The average economy for a variable load

will not necessarily be the same as the economy at the average load, and may differ considerably from it. The computations presented in the paper, however, are designed, as stated, more to define the problem than to solve it, wherefore only one case is worked out.

Inasmuch as the figures are presented for the Connors Creek plant as it stands, no consideration has been given to the use of economizers.

Mr. Pigott asks why the systems described were used, and Mr. Scott asks whether like systems would be installed again. This may perhaps best be answered by stating that we are now designing a new station, and that it will comprise an auxiliary power and feedwater heating system differing from the Connors Creek scheme only in details which make for steadier operation and readier control. The Connors Creek system is shortly to be altered along similar lines. A system of this type we believe presents maximum flexibility, maximum possibilities for thermal economy, and ample reliability.

The ultimate goal of power-station design is obviously the minimum cost of power output consistent with reliable service. In recent years rapid progress has been made toward this goal, and we seem to have reached a stage in the journey where even small advances are to be accomplished only by strenuous efforts and radical changes in practice. In the turbine room, we must give attention to such things as the Ljungstrom turbine, the reheating turbine approximated by Ferranti, and the mercury-vapor turbine. The bleeding of steam from the main turbines for heating boiler feedwater seems to open the way for important developments. In the boiler room we are faced with such possibilities as pulverized coal, water backs and screens, air pre-heaters, and the more general use of economizers. The arrangement of auxiliary systems will of course change with changes in the major equipment, but so long as we use steam the object will ever be the generation of the maximum power from the greatest possible steam flow which can be condensed by the boiler feedwater.

What the power plant of twenty or thirty years hence will be, none can tell. That it will be more economical than our present equipments seems certain. That it will resemble them physically is open to serious question.

J. H. Lawrence. In designing the Hell Gate station the subject of high pressure was investigated and it was the consensus of opinion of the engineers that it was not advisable to design the

station for over 250 lb. pressure at the turbine and a maximum temperature of 625 deg. fahr. It was felt that an improvement in economy would not compensate for the extra expense and troubles which would result.

Local ordinances in New York City prohibit the storage of a sufficient quantity of oil to insure continuous operation of a large station.

Duplex drive for auxiliary should cause no trouble where properly handled. The load should not be entirely taken off the turbine.

It would be difficult to prepare a standard heat balance code, although the result would be extremely valuable if it could be put in workable form. It was proposed that the four papers on heat balance be prepared on a hypothetical design for a station having 150,000 kw. maximum load, but each author was to work up the heat balance in accordance with the system in use at his particular station. However, it was pointed out that the results, while comparable, would be only theoretical and hence of less value than actual operating figures.

One of the principal features of the auxiliary layout at Hell Gate station is the control system. All auxiliaries, after being brought to speed, are under the direction of the station auxiliary operator located in the turbine room. All auxiliaries are automatically controlled except the forced and induced draft fans, which are controlled by the station auxiliary operator from his post in the turbine room. The operator has instruments for each auxiliary in the station on a board where he is stationed.

The writer agrees with Mr. Orrok that a station designed for steam auxiliaries and operating at a certain load factor would show results but slightly different from a plant with electric auxiliaries. However, in a public service station the load factor is not easily governed. It must be accepted as it exists and varies. Under these conditions the electric auxiliary station will show sufficiently better results to warrant its adoption over steam auxiliaries.

One point to consider with regard to small turbine drive is the high temperature of the exhaust steam, — often as high as 500 deg. fahr. Small auxiliaries are not designed ordinarily for this high temperature and as a result the casings are very often distorted.

Mr. Pigott is correct in saying that the heat balance must be compiled on a variety of loads in order to determine the best method of driving auxiliaries. In other words, the load curve must be studied and the auxiliary system adapted to the same.

In answer to Mr. Pope's criticism of main turbine running away when supplied with low-pressure steam at light loads, it may be pointed out that the units at Hell Gate station are equipped with valves which close with the main turbine valve, eliminating all danger of over-speeding.

Some reference has been made to the temperature at which the best efficiency is obtained by preheating the feedwater before entering the boiler. It has been stated that this temperature is in the neighborhood of 150 deg. fahr. I do not agree with this statement, as stage heating is one which results in higher feedwater temperature and higher efficiency than heating water to low temperature.

W. M. KEENAN. It was rather disappointing to find that the discussion of the heat-balance papers brought forth no other methods of treating the problem and not very much direct criticism of those presented. Although disappointing, the explanation probably is that the treatment of a power plant as a single composite heat engine, with the turbo-generators, auxiliaries, and the boilers considered merely as component parts of the whole, has not had, as yet, a very wide presentation, and the general attitude toward the subject seems to be one rather of expectancy than direct knowledge. In each case the number of variables is large and the absence of a standardized method of calculation renders difficult a comprehensive interpretation of the results arrived at by anyone unfamiliar with the particular line of reasoning pursued.

The metallurgical aspect of the heat-balance problem received passing consideration in the discussion, yet it is one of tremendous importance to future design.

The lack of proper materials to meet conditions with which the power-plant engineer has to deal is evidenced at many points outside the turbines, generators and condensers whose requirements up to the present time have received most attention.

If there were a material that could be used for economizer tubes, of such nature that water at a temperature as low as 70 deg. fahr. could be put in the economizer, with no bad effect either from entrained air or sweating, a very simple plant could be designed having a very high overall efficiency. A combination boiler-economizer could be used as practically the sole heat supplying apparatus in the plant, the condensate and makeup passing directly to the economizer without the complication of heaters or dioxidizers of any sort. The auxiliaries would be motor driven and the reliability of

auxiliary drive secured by means other than the use of house generators.

There are many other instances in which materials having different properties than those ordinarily employed could be used to advantage; for example, consider the investment saving and the simplification resulting from the use of a material for steam lines that would be suitable for high temperatures and having a zero coefficient of expansion or one of negligible value.

No. 1804

ON THE ART OF MILLING

By John Airey, Ann Arbor, Mich.
Member of the Society
and
Carl J. Oxford, Detroit, Mich.
Associate-Member of the Society

This paper gives particulars of an investigation undertaken at the University of Michigan for the purpose of finding a rational basis for the action of a milling cutter.

It is shown that metal is removed more efficiently with thick chips than with thin chips. It follows from this that, other conditions being equal, including speed and feed per minute, the cutter with the fewest teeth gives the greatest efficiency. However, it is evident that the efficiencies of two cutters with different numbers of teeth are equal provided the table feeds be adjusted so that the same feed per tooth is effected. This gives a definite working theory on the influence of spacing.

It is definitely established that for a given material, tooth shape and sharpness, thickness of chip is the sole criterion of the efficiency with which metal is removed in milling and that increase of spacing over that required for free cutting is a handicap. Present-day high-powered cutters have several times the chip space needed. Limitation of machine power has doubtlessly been the chief factor in giving a false bias to the influence of spacing.

Formulas for determining the number of teeth for a known diameter of cutter and for determining the depth are included in the paper, as well as a geometrical construction for obtaining the best shape of tooth.

THE industrial value of scientific analysis applied to basic shop processes appears to be little appreciated when that analysis is contrasted with investigations carried on to improve the design of the manufactured product. This is explained partly by the cleavage between the engineering department and the operating department in almost all plants. It is also partially explained by shrinkage in the significance of the term "engineering" until it means "design" only when used in a quantity-production atmosphere. This is either the cause or the effect of our engineering schools' keeping out of touch with production proper, and consequently research

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energy is lavished on all features of design, each minor side issue usually having its full quota of devotees.

2 In contrast to this, research endeavor covering basic shop processes appears to be badly neglected. Taylor, in this country, and Nicolson, in England, have done classical work in the field of lathe-tool action. There is much to be done and many fundamental shop processes are yet untouched. A plea is here made that our engineering schools devote the same intensity of attention to shop processes that they now devote to design topics. The economic results to industry would be greater though not so spectacular. If a hole can be drilled more quickly or a surface can be milled more rapidly, then the cost of practically all classes of finished products is affected. This is true of improvements in all basic shop processes, whereas improvements in design benefit only a limited field — just one class of finished product. The economy accruing from improvements in shop processes being distributed over a larger territory makes it less spectacular, and this, indirectly, might account for the decreased attention given it.

3 The practice of milling has, during the present century and the last decade of the ninteenth century, encroached decidedly on the field occupied by the shaper and planer. The action of milling possesses advantages over shaping; it also has its drawbacks. Much work must be done before we can state the relative merits or potentialities.

4 In the art of milling, the only published investigations have been handled by machine-tool builders. Attention has in consequence been focused more on the machine than on the cutter. Further, in the investigations already made, the objects sought have usually had the defect of being too immediate. This paper describes an attempt to investigate the fundamental principles underlying the action of milling.

5 It is shown in the present investigation that metal is removed more efficiently with thick chips than with thin chips. It follows from this that, other conditions being equal, including speed and feed per minute, the cutter with the fewest teeth gives the greatest efficiency. However, it is evident that the efficiencies of two cutters with different numbers of teeth are equal provided the table feeds be adjusted so that the same feed per tooth is effected. This gives a definite working theory on the influence of spacing.

6 In addition to the present investigation, experiments published in substantiation of the advantage of wide spacing agree in

confirming this theory. It is definitely established that for a given material, tooth shape and sharpness, thickness of chip is the sole criterion of the efficiency with which metal is removed in milling and that increase of spacing over that required for free cutting is a handicap. Present-day high-powered cutters have several times the chip space needed. Limitation of machine power has doubtlessly been the chief factor in giving a false bias to the influence of spacing.

7 The work described in this paper started in May, 1920, through a desire of the National Twist Drill and Tool Company of Detroit to find a rational basis for the action of a milling cutter so far as this could be removed from the region of empiricism. This company furnished the financial support throughout. The experiments have been conducted in the Engineering Shops of the University of Michigan, at the National Twist Drill and Tool Company, the Lincoln Motor Company, and the Hudson Motor Car Company. The greater part of the work was done at the University of Michigan, viz., all experiments on the Chip Investigator and on the Thrust and Torque Measurer. The work on the Machineability Tester was done at the National Twist Drill and Tool Company and the tests on endurance were run at the two automobile plants.

8 The writers take this opportunity to express their appreciation of the encouragement given throughout by Mr. Howard L. McGregor, of the National Twist Drill and Tool Company. Thanks are accorded to Mr. H. M. Leland, who willingly made the facilities of the Lincoln Motor Company available, and to Mr. Roy D. Chapin, who similarly opened the extensive Hudson plant. Acknowledgment is also made of the unfailing courteous coöperation given by Mr. R. L. Welborn of the Lincoln Motor Company and by Mr. I. B. Swegles of the Hudson Company. Lastly, acknowledgment is made of the invaluable assistance rendered throughout by Mr. Ingle B. Whinery, on whom the brunt of the work fell. Mr. Whinery was connected with this work from the start: first as a graduate student at the University of Michigan and later, throughout the year 1920–21, as holder of a National Small Tool Fellowship (awarded by the National Twist Drill and Tool Company), and finally as an engineer with the company. Mr Whinery ably assisted in the troublesome development stage of equipment, later conducted most of the experiments personally and finally prepared all the data for this paper.

A DYNAMICAL STUDY OF SINGLE-TOOTH ACTION

9 The chip taken by a plain slab mill is very different from the chip taken by a lathe or shaper tool. A milling chip is shown exaggerated in the shaded portion of Fig. 1. The two circles show the outline of the cutter at the beginning and at the end of the interval taken for the cutter to revolve through the angle subtended by one tooth. The shaded portion will therefore represent the material removed by one tooth and will have an area $f \times d$.

10 The chip starts infinitely thin and its thickness gradually increases to a maximum just before the finish, from which point it quickly decreases. This decrease is practically instantaneous in an unexaggerated chip. The question arises, how does the force vary throughout the cutting of this chip?

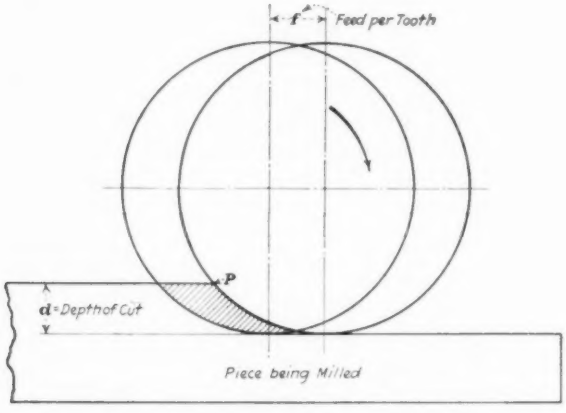


FIG. 1 SHAPE OF MILLING CHIP

11 The first problem is to find how the tangential force varies in relation to chip thickness. For measuring this force while the chip is being formed under reasonable commercial velocity conditions, a dynamical measuring method is the only solution, as a chip is completely cut in about one tenth of a second.

12 *Description of Chip Investigator.* Fig. 2 is a diagrammatic representation of the apparatus here known as the Chip Investigator. A block of material is shown bolted to a milling machine table and a single-tooth cutter in place, the latter being rigidly attached to the sector-shaped arm. This arm can partially rotate

about an axis at its lower end, and in so doing it carries the cutter through an arc long enough to make a chip of the desired length.

13 If the arm be allowed to fall freely, the angular velocity will increase throughout the fall. If this be repeated after the table is adjusted so as to cause a cut to be taken, then the next fall of the arm will be retarded, because it must give up sufficient energy to do the cutting. Weights can be placed on the Pad for Additional

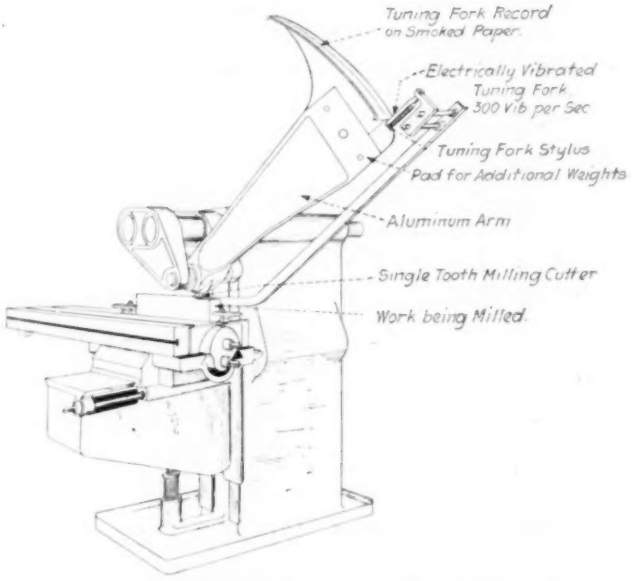


FIG. 2 DIAGRAM OF CHIP INVESTIGATOR

Weights (see Fig. 2) sufficient to secure a roughly uniform speed during cutting.

14 The investigation of the velocity at each instant is made by means of a tuning fork, which gives a record on smoked paper. The outer surface of the arm, remote from the axis of oscillation, is concentric: to this a glazed paper ribbon is attached and then smoked by a gas flame.

15 An angle iron bolted to the front face of the machine carries at its upper end a Koenig electrically driven tuning fork making 500 vibrations per second. As the arm falls a wavy line is traced on the smoked paper by a stylus attached to one leg of the tuning fork. Each wave represents a definite interval of time, viz., two

one-thousandths of a second. The smoked-paper record is immediately dipped in thin shellac varnish and hung up to dry.

16 The analysis and computation of smoked-paper records is given fully in Appendix No. 1. An example of a chip study is given in Fig. 3, where it is very clearly shown that the force employed is not proportional to the chip thickness. In other words, material is removed more efficiently as the chip becomes thicker. This is confirmed later by experiments on varying feeds and furnishes the true explanation of the supposed advantages of coarse-tooth cutters.

17 Over twelve hundred chips were cut and the energy computed. These were taken at three different cutting speeds, viz.,

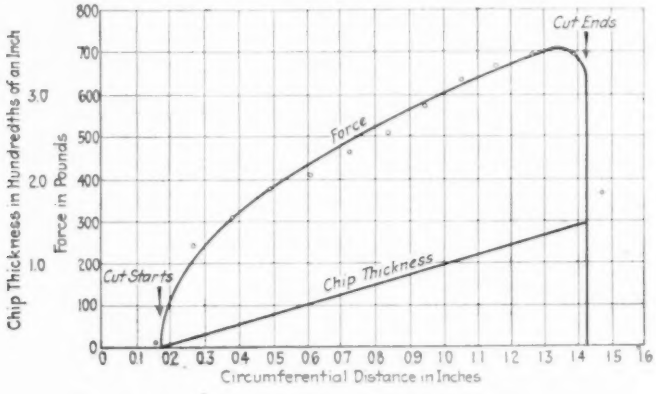


FIG. 3 FORCE–SPACE CURVE OF CHIP FORMATION

17.5, 32 and 44 ft. per min., respectively. The average of the energy required at different speeds was remarkably equal. If the energy used at lowest speed be taken as 100, the results are:

Speed in ft. per min.....	17.5	32	44
Energy	100	101.8	97.4

As there appears no law in this and the variation is slight, presumably due to experimental error, the conclusion was reached that speed does not influence energy required.

18 *Influence of Rake and Feed on Cast Iron.* The results of the analysis of sixteen sets of chip formations in cast iron are given in Table 1 and shown graphically in Figs. 4 and 5. Each set consists of three chips formed at the different speeds as stated in Par. 17.

19 The area under each curve gives a numerical representation of the average production of each tool over a certain region of chip weight. Taking the 0-deg. rake tool as 100 we have:

Production of 0-deg. rake tool = 100
Production of 10-deg. rake tool = 133
Production of 20-deg. rake tool = 138.4
Production of 30-deg. rake tool = 142.1

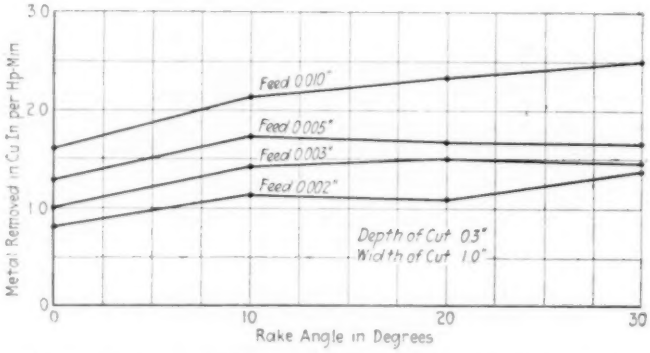


FIG. 4 CAST IRON: METAL REMOVED AS FUNCTION OF RAKE

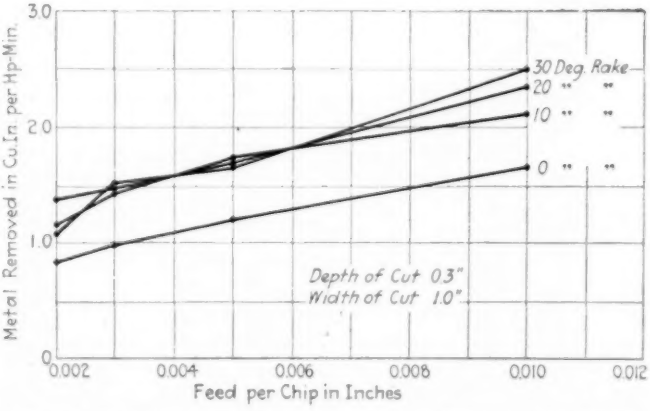


FIG. 5 CAST IRON: METAL REMOVED AS FUNCTION OF FEED

20 *Influence of Rake and Feed on Bronze.* The results of the analysis of sixteen sets of chip formations in bronze are given in the following table. As in the last section the consistent superior

economy of heavy feeds is observed. This information is condensed as before and results in:

Production of 0-deg. rake tool = 100
Production of 10-deg. rake tool = 113.2
Production of 20-deg. rake tool = 123
Production of 30-deg. rake tool = 118

TABLE 1 INFLUENCE OF RAKE AND FEED IN CUTTING CAST IRON

Clearance angle, 2 deg.; spiral angle, 0 deg.; depth of cut, 0.3 in.; width of cut, 1 in.

Experiment No.	Rake angle, deg.	Feed in inches	Energy in ft-lb.	Cu. in. per hp-min.
52	0	0.002	24.2	0.818
53	0	0.003	30.4	0.977
54	0	0.005	38.3	1.293
55	0	0.010	60.8	1.627
59	10	0.002	17.3	1.143
60	10	0.003	21.1	1.407
61	10	0.005	28.9	1.712
62	10	0.010	47.1	2.105
66	20	0.002	18.1	1.093
67	20	0.003	19.8	1.500
68	20	0.005	29.3	1.690
69	20	0.010	42.7	2.319
73	30	0.002	14.2	1.394
74	30	0.003	20.6	1.441
75	30	0.005	29.7	1.666
76	30	0.010	39.6	2.500

TABLE 2 INFLUENCE OF RAKE AND FEED IN CUTTING MACHINE STEEL

Clearance angle, 3 deg.; spiral angle, 0 deg.; depth of cut, 0.3 in.; width of cut, 1.0 in.

Experiment No.	Rake angle, deg.	Feed in inches	Energy in ft-lb.	Cu. in. per hp-min.
250	0	0.0025	45.1	0.549
251	0	0.0050	72.8	0.680
252	0	0.0075	94.3	0.787
253	0	0.0100	103.1	0.960
289	10	0.0025	38.8	0.638
290	10	0.0050	60.0	0.825
291	10	0.0075	76.8	0.967
292	10	0.0100	94.4	1.049
260	20	0.0025	25.0	0.990
261	20	0.0050	46.3	1.068
262	20	0.0075	59.9	1.240
263	20	0.0100	71.4	1.386
265	30	0.0025	25.9	0.955
266	30	0.0050	41.2	1.200
267	30	0.0075	55.0	1.350
268	30	0.0100	62.5	1.584

21 *Influence of Rake and Feed on Machine Steel.* The results of the analysis of sixteen sets of chip formations in machine steel are given in Table 2 and presented graphically in Figs. 6 and 7.

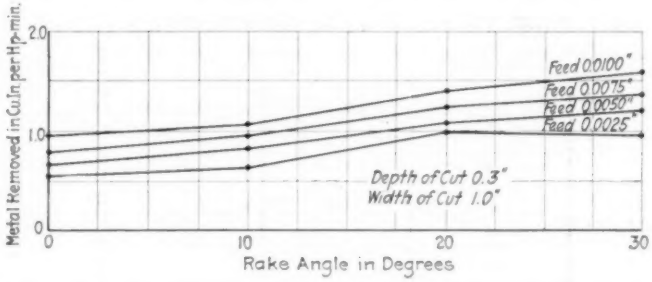


FIG. 6 MACHINE STEEL: METAL REMOVED AS FUNCTION OF RAKE

The consistent superiority of heavy feeds is again noted. Again condensing the above information:

Production of 0-deg. rake tool = 100
Production of 10-deg. rake tool = 118.7
Production of 20-deg. rake tool = 157.2
Production of 30-deg. rake tool = 172

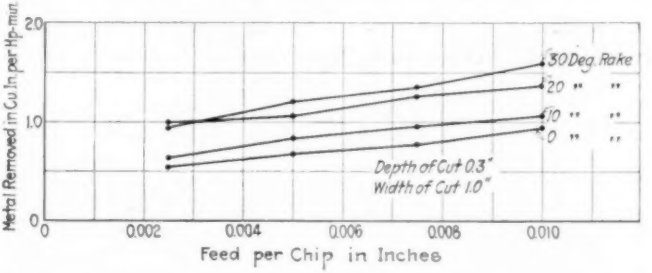


FIG. 7 MACHINE STEEL: METAL REMOVED AS FUNCTION OF FEED

22 *Influence of Rake and Feed on Carbon Tool Steel.* The results of the analysis of sixteen sets of chip formations in tool steel are given in Table 3, and graphically in Figs. 8 and 9. Again condensing the above information:

Production of 0-deg. rake tool = 100
Production of 10-deg. rake tool = 106
Production of 20-deg. rake tool = 112.2
Production of 30-deg. rake tool = 112.0

23 *Effect of Lubrication.* Various experiments were run with lubricant. The results must be taken qualitatively rather than quantitatively.

TABLE 3 INFLUENCE OF RAKE AND FEED IN CUTTING TOOL STEEL

Clearance angle, 3 deg.; spiral angle, 0 deg.; depth of cut, 0.3 in.; width of cut, 1.0 in.

Experiment No.	Rake angle, deg.	Feed in inches	Energy in ft-lb.	Cu. in. per hp-min.
331	0	0.0025	39.6	0.625
332	0	0.0050	64.0	0.774
333	0	0.0075	89.5	0.830
334	0	0.0100	102.8	0.960
336	10	0.0025	43.1	0.575
337	10	0.0050	60.7	0.820
338	10	0.0075	79.5	0.924
339	10	0.0100	97.6	1.012
341	20	0.0025	38.8	0.639
342	20	0.0050	60.4	0.819
343	20	0.0075	73.7	1.005
344	20	0.0100	90.0	1.100
346	30	0.0025	37.6	0.659
347	30	0.0050	59.3	0.832
348	30	0.0075	76.1	0.972
349	30	0.0100	90.9	1.095

24 The action of lubrication in milling appears to be different from its action in lathe work. In lathe action, lubrication proper of the cutting tool is probably non-existent because of the unlike-

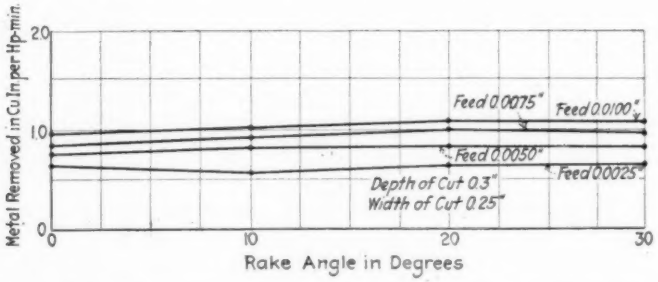


FIG. 8 TOOL STEEL: METAL REMOVED AS FUNCTION OF RAKE

lihood of the lubricant finding its way to the point of the tool during cutting. The result in the lubricant being useful in lathe work only in proportion to its heat-carrying capacity and to its freedom from properties injurious to the mechanical condition of the machine.

25 In milling there is a decided influence entirely apart from heat-conducting properties. If the surface exposed by the last chip was very slightly smeared with an oiled finger (the oil film was scarcely discernible), the energy consumed in taking the next chip was from 10 per cent to 25 per cent lower in nearly all cases.

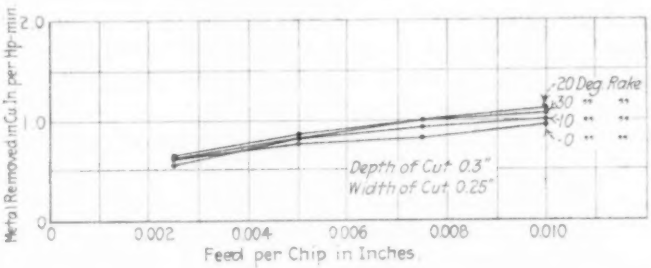


FIG. 9 TOOL STEEL: METAL REMOVED AS FUNCTION OF FEED

26 On plotting batches of results, however, the surprising situation was found of oil proving advantageous with low rakes in bronze and detrimental with high rakes. The existence of some unknown varying factor was suspected and the result would have been rejected and forgotten but for the fact that the same effect showed up in experiments with machine steel made at a different time. The metal removed per horsepower-minute is shown plotted against rake for the two materials in Fig. 10.

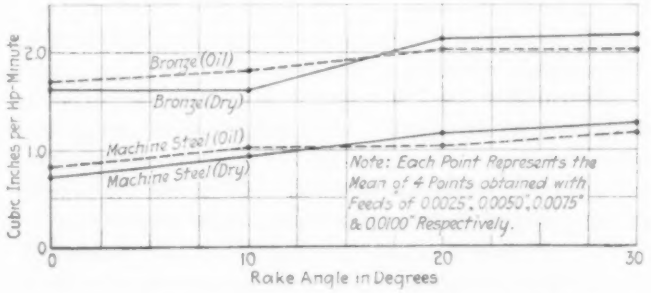


FIG. 10 EFFECT OF OIL ON MACHINE STEEL AND BRONZE

27 The authors hesitate to accept the foregoing unreservedly until further investigation sheds some light on its rationality. A study of the resultant force during the formation of a chip with

reference to its changing magnitude and *direction* would give the needed explanation. This has not yet been undertaken.

28 *Effect of Chatter.* When chatter occurs, violent fluctuation of energy results. The energy consumed was usually greater than with smooth cutting. The authors believe that, due to chatter alone, *increase* of energy invariably results and that, where a decrease appeared, it was caused by the cut being lighter than supposed to be. This can be brought about by the chatter causing an increase of force normal to the work and this in turn springing the work or cutter.

29 *Effect of Clearance.* Clearance was found not to affect energy consumed. Chatter, wear of tool and liability of tool to snip are affected by clearance.

30 *Method of Determining Machineability.* The word "machineability" is suggested to indicate the ease with which a given material can be machined. This of course presupposes standardized conditions of tool and size and shape of chip. If all conditions be standardized except the quality of the material being cut, then the machineability can be expressed in foot-pounds of energy required to remove one cubic inch of the material under investigation.

31 It is well known that machineability is a continual bone of contention in machining departments and that no accepted satisfactory way has yet been devised for its measurement. It is generally admitted that even though chemical analysis, tensile strength, elongation, Brinell hardness number and scleroscope hardness number are known, yet the machineability is an unknown factor. In fact, the War Department gave to the American Research Council as their major topic the development of some definite means of determining machineability.

32 A machine has been developed along these lines and is described later in the paper. For identification purposes this is referred to as the "Machineability Tester." Experiments with this machine are presented at the end of the paper rather than here, because they were the last experiments to be performed.

33 To summarize: The authors had, at this state of development, no definite idea leading to a rational structure which would connect up the results. They regard the work in this section as useful in opening up territory, thereby enabling them to formulate more definite layouts for subsequent work, and believe that broadly the following conclusions are justified:

(*a*) As a chip thickens, metal is removed more easily per unit volume

(*b*) As rake angle is increased, metal is removed more easily per unit volume. Also this improvement is continuous but not uniform

(*c*) The advantage of rake depends on the material and is more pronounced with ductile materials

(*d*) The effect of lubrication proper (apart from cooling action) is decided, but the information available is inadequate to formulate it.

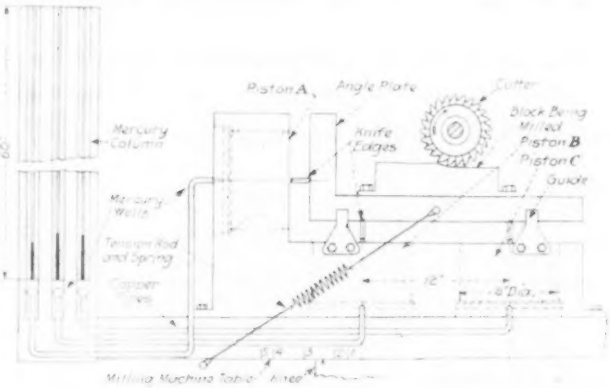


FIG. 11 DIAGRAM OF THRUST AND TORQUE MEASURER

COMPLETE FORCE ANALYSIS OF CUTTER IN ACTION

34 Numerous experiments on power consumption have been made by different investigators, but the method employed has been to measure the electrical input required to drive the machine. This confuses the unknown action of the cutter with the unknown machine loss and has therefore been rejected for the present investigation.

35 *Features to be Measured.* It was thought desirable to measure both the vertical and the horizontal components of the force exerted by the cutter. This necessitated developing an instrument that would measure both force and location, i.e., in all, three quantities: magnitude, direction and location. Or, which amounts to the same thing, the two components and the location may be measured.

36 *Description of Equipment.* The apparatus developed for force and power analysis is shown diagrammatically in Fig. 11, and is designated the "Thrust and Torque Measurer."

37 The angle plate, to which is bolted the material being

cut, is carried on three knife edges, through which the forces are transmitted to pistons A, B and C. The double-knife-edge principle used transmits only a horizontal force to piston A and only vertical forces to pistons B and C.

38 The fluid pressure under the pistons is transmitted through pipes and measured by means of mercury columns shown at the left of Fig. 11. As the table travels in feeding, the location of the downward thrust of the cutter in relation to cylinders B and C changes continuously. The table traveling from left to right results

TABLE 4 INFLUENCE OF RAKE IN CUTTING MACHINE STEEL

Number of teeth, 34; diameter, 6 in.; clearance angle, 5 deg.; spiral angle, 0 deg.; depth of cut, 0.3 in.; width of cut, 0.5 in.

Exp. No.	Date	Rake angle, deg.	Feed, in. per min.	Speed, r.p.m.	Vol. per tooth, cu. in.	Torque, in.-lb.	Cu. in. per hp-min.
1001	2-23-21	0	0.92	27.9	0.000145	693	0.448
1002	2-23-21	0	2.36	27.9	0.000375	1390	0.585
1003b	2-23-21	0	3.10	19.3	0.000710	2050	0.748
1004a	2-24-21	10	0.92	27.9	0.000145	455	0.685
1005	2-24-21	10	2.36	27.9	0.000375	980	0.830
1006	2-24-21	10	3.10	19.3	0.000710	1710	0.896
1007	2-25-21	10	0.92	27.9	0.000145	488	0.637
1008	2-25-21	10	2.36	27.9	0.000375	1040	0.781
1010	2-29-21	0	0.92	27.9	0.000145	618	0.504
1011	2-29-21	0	2.36	27.9	0.000375	1260	0.645
1012	2-29-21	0	3.10	19.3	0.000710	1954	0.783
1013	3- 7-21	20	0.92	27.9	0.000145	291	1.068
1014	3- 7-21	20	2.36	27.9	0.000375	740	1.100
1015	3- 7-21	20	3.10	19.3	0.000710	1432	1.070

in the pressure decreasing on C and increasing on B. The readings of mercury columns must therefore be taken for known definite locations of table. Five marks were made in the table, and as these successively came into alignment with a zero mark on the saddle, the readings were taken.

39 *Influence of Rake on Machine Steel.* The results of fourteen experiments are shown in Table 4 and graphically in Fig. 12. The noticeable consistency in the spacing of the pairs of values given as results of the zero-rake cutter is due to the second three having been taken in a different part of the block from the first three. The results plotted in Fig. 12 show the decided advantage of rake. The advantage of heavy chips is also clearly shown except in the case of the 20-deg. tool.

40 The authors do not draw any conclusion regarding this exceptional case as the experiments available are too few. In general there is conclusive evidence that heavy chips are more economical of power than small chips, within the practical limits of present-day milling practice. The exact extent of this economy cannot be stated at the present state of the art. It is certain, however, that this condition is a function both of the material and of the rake. The influence of rake appears to depend largely on ductility.

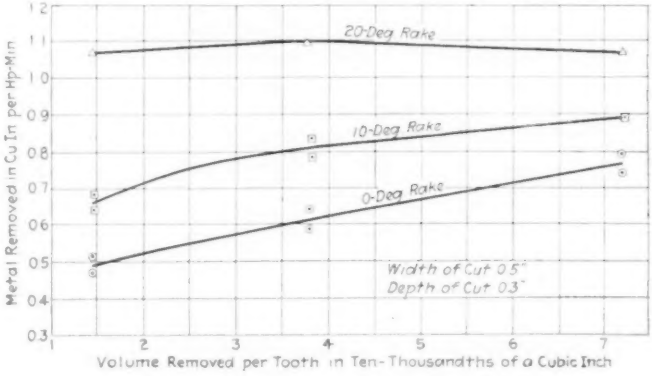


FIG. 12 INFLUENCE OF RAKE IN CUTTING MACHINE STEEL

41 The area under each curve gives a numerical representation of the average production of each tool over a certain region of chip weight. Taking the 0-deg. rake tool as 100 we have:

Production of 0-deg. rake tool = 100
Production of 10-deg. rake tool = 130
Production of 20-deg. rake tool = 176

42 *Influence of Rake on S.A.E. 3240 Alloy Steel.* The results of twelve experiments are shown in Table 5 and in Fig. 13. Again taking the area under the curve to represent the productivity, we obtain:

Production of 0-deg. rake tool = 100
Production of 10-deg. rake tool = 130.5
Production of 20-deg. rake tool = 137.5

In this case the advantage of rake is not so decided as with the more ductile machine steel — particularly above 10 deg. In addition, the same tendency is observed to reduce the advantage of thicker chips with a high rake.

43 *Influence of Depth of Cut.* Thirty-five experiments were run with various depths of cut, various feeds, and two different rakes. Analysis of these results gave rise to a belief that the block of material used was far from homogeneous. These experiments are not recorded in detail for this reason. It is advisable, however, to record certain impressions obtained and tentative conclusions.

44 If material removed per hp-min. be plotted on a chip-volume base, then a shallow cut gives a curve higher than that of a deeper cut. This is not surprising. It has been shown that, as

TABLE 5 INFLUENCE OF RAKE IN CUTTING ALLOY STEEL

Number of teeth, 34; diameter, 6 in.; clearance angle, 5 deg.; spiral angle, 0 deg.; depth of cut, 0.3 in.; width of cut, 0.5 in.

Exp. No.	Date	Rake angle, deg.	Feed, in. per min.	Speed, r.p.m.	Vol. per tooth, cu. in.	Torque, in-lb.	Cu in. per hp-min.
1017	3- 8-21	20	0.92	27.9	0.000145	296	1.048
1018	3- 8-21	20	2.36	27.9	0.000375	818	0.994
1019	3- 8-21	20	3.10	19.3	0.000710	1340	1.142
1020	3- 9-21	10	0.92	27.9	0.000145	334	0.930
1021	3- 9-21	10	2.36	27.9	0.000375	782	1.040
1022	3- 9-21	10	3.10	19.3	0.000710	1392	1.100
1023	3-10-21	10	0.92	27.9	0.000145	363	0.856
1024	3-10-21	10	2.36	27.9	0.000375	803	1.010
1025	3-10-21	10	3.10	19.3	0.000710	1505	1.020
1029	3-15-21	0	0.92	27.9	0.000145	429	0.706
1030	3-15-21	0	2.36	27.9	0.000375	1045	0.778
1031	3-15-21	0	3.10	19.3	0.000710	1930	0.795

a chip thickens, the material is removed with less energy per unit volume. If two chips of equal volume but of different depths of cut be taken, then the chip of the shallow cut is shorter and thicker.

45 It appeared probable that, with a given cutter and uniform material, if the feed per tooth and depth of cut were varied, the number of cubic inches per hp-min. would prove to be a definite function of maximum chip thickness. Experimental verification of this is given later in the paper.

46 *Influence of Lubricant.* The results of six experiments, which merely touched the fringe of the effect of lubrication, are given in Table 6. It is dangerous to generalize too broadly from a few results, but one can at least conclude that lubrication effects a reduction of energy and that this reduction appears more pronounced in light cuts. It should be again noticed that this is al-

most pure lubricating action and that the conduction of heat by the oil plays only an insignificant part.

47 *Effect of Spiral.* A 34-tooth non-spiral cutter, 6 in. in diameter and having a 10-deg. rake, was run in comparison with cutters similar in all respects excepting that they possessed a $12\frac{1}{2}$-deg.

TABLE 6 EFFECT ON LUBRICATION

Number of teeth, 34; diameter, 6 in.; clearance angle, 5 deg.; spiral angle, 0 deg.; depth of cut, 0.3 in., width of cut, 0.5 in.

Exp. No.	Date	Rake angle deg.	feed, in. per min.	Speed, r.p.m.	Vol. per tooth cu. in.	Torque, in-lb.	Cu. in. per hp-min.
.1026 Oil	3-10-21	10	0.92	27.9	0.000145	306	1.018
1026 Dry	3-10-21	10	0.92	27.9	0.000145	364	0.855
1027 Oil	3-10-21	10	2.36	27.9	0.000375	760	1.070
1027 Dry	3-10-21	10	2.36	27.9	0.000375	875	0.930
1028 Oil	3-10-21	10	3.10	19.3	0.000710	1335	1.148
1028 Dry	3-10-21	10	3.10	19.3	0.000710	1430	1.071

spiral angle. Two cutters with left- and right-hand spirals, respectively, were used in combination, thus eliminating end thrust. Two different combinations of speeds and feeds were used. The material cut was machine steel. The results were as follows:

R.p.m.	Feed, in. per min.	Cubic Inches Per Hp-min.		Lost output of spiral cutter. per cent
		0-deg. spiral	12½-deg. spiral	
28.1	0.92	0.422	0.338	20
16.1	0.92	0.568	0.458	19.4

48 The authors are well aware that a general belief exists in the superior power efficiency of spiral cutters. Frequent statements to this effect can be found in the literature of the subject. In this investigation it has been impossible to substantiate these. It is difficult to understand the growth of the belief that spiral cutters are more efficient, because abstract analysis leads to the conclusion that spiral cutters must be inferior to straight cutters. By "inferior" the meaning intended is in reference to power efficiency only. The use of a spiral angle is heartily recommended, for it results in continuity of action, tends to avoid chatter and keeps the driving power

more smoothly constant. The smallest angle consistent with smoothness of action should be used.

49 *Theory of Tooth Spacing.* Opinions differ on the influence of tooth spacing or, in other words, the relative merits of coarse-tooth compared with fine-tooth cutters. Many cutter manufacturers standardize two types, the fine-tooth cutter and the "high-powered" cutter. Experiments have been published purporting to show the advantage of coarse-tooth cutters. Possibly many others have been made and have probably convinced the experimenter that the claims made were justified.

50 There is an interesting delusion in this subject and it is

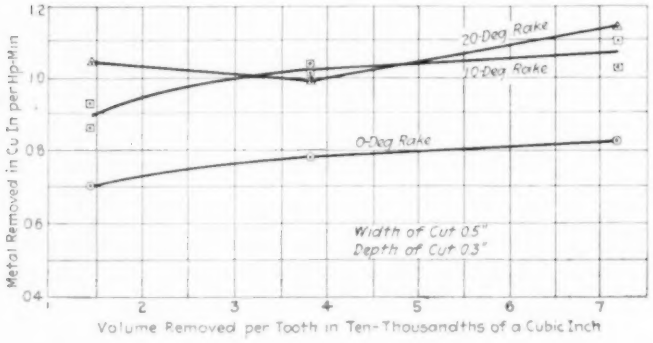


FIG. 13 INFLUENCE OF RAKE IN CUTTING ALLOY STEEL

unfortunate that it has held sway for so long, for it has resulted in inefficient types of cutters being used. Most of the so-called high-powered cutters today on the market would be improved if more teeth were given.

51 It has been demonstrated earlier that the force required to remove metal does not increase in proportion to the chip thickness. It follows directly from this (and has been independently demonstrated by the Thrust and Torque Measurer) that as feed is increased the force will not increase in proportion.

52 Another way of stating the last paragraph is that the heavier the chip, the more metal removed per hp-min. If one were to make a comparison of two cutters with different numbers of teeth but run at the same speed and feed per minute, then obviously the cutter with fewer teeth would remove the metal with less power *because the chips are thicker,* there being fewer teeth cutting per

minute for the same amount of metal removed. It appears illogical, a priori, to suppose that the force action of a single tooth under any given set of conditions should be influenced by the location of neighboring teeth, *if free cutting is in operation.* This view is fully substantiated by experiment.

53 Referring to the preceding paragraph, if the feed has been increased with the fine-tooth cutter in the ratio of the number of teeth in the two cutters (thereby making a chip of the same thickness as with the coarse-tooth cutter), *then exactly the same number of cubic inches per hp-min. would have been removed as were removed*

TABLE 7 INFLUENCE ON TOOTH SPACING

Exp. No.	Date	Material	No. of teeth	R.p.m.	Feed, in. per min.	Vol. of chip, cu. in.	Torque, in-lb.	Cu. in. per hp-min.
1110	7-8-21	Alloy St.	26	33.7	0.92	0.00016	840	0.306
1111	7-8-21	Alloy St.	26	33.7	2.36	0.00040	1600	0.412
1112	7-8-21	Alloy St.	26	19.3	2.36	0.00071	2360	0.487
1116	7-20-21	Alloy St.	34	33.7	0.92	0.00012	720	0.357
1117	7-20-21	Alloy St.	34	33.7	2.36	0.00031	1400	0.472
1118	7-20-21	Alloy St.	34	19.3	2.36	0.00054	2300	0.500
1119	7-20-21	Alloy St.	18	33.7	0.92	0.00023	610	0.421
1120	7-21-21	Alloy St.	18	33.7	2.36	0.00058	1250	0.528
1121	7-21-21	Alloy St.	18	19.3	2.36	0.00102	1930	0.595
1122	7-28-21	Mach. St.	18	33.7	0.92	0.00023	550	0.467
1123	7-28-21	Mach. St.	18	33.7	2.36	0.00058	1040	0.636
1125	8-2-21	Mach. St.	34	33.7	0.92	0.00012	610	0.422
1126	8-2-21	Mach. St.	34	33.7	2.36	0.00031	1310	0.504
1128	8-2-21	Mach. St.	34	19.3	2.36	0.00054	1850	0.621
1129	8-4-21	Mach. St.	26	33.7	0.92	0.00016	650	0.396
1131	8-4-21	Mach. St.	26	23.3	1.23	0.00030	1210	0.412
1132	8-4-21	Mach. St.	26	19.3	2.36	0.00071	1970	0.584
1133	8-5-21	Mach. St.	18	57.1	0.92	0.00013	350	0.434
1134	8-5-21	Mach. St.	18	27.8	1.23	0.00037	800	0.521
1136	8-5-21	Mach. St.	26	33.7	0.92	0.00016	590	0.435
1137	8-6-21	Mach. St.	26	33.7	2.36	0.00040	1190	0.555
1138	8-6-21	Mach. St.	26	19.3	2.36	0.00071	1710	0.672
1139	8-6-21	Mach. St.	26	23.3	2.36	0.00058	1560	0.615

with the coarse-tooth cutter, and more actual cubic inches of material would have been removed per minute, provided the limit of machine power had not been reached.

54 If comparative tests had always been analyzed from the point of view of *chip volume* (or maximum chip thickness, if the depth of cut varies), then the unmerited virtue of wide spacing would never have been recognized.

55 Three 6-in. diameter side mills were taken, 0.5 in. thick

and with 10 deg. rake. They had respectively 34, 26 and 18 teeth. Various tests were run in alloy steel and in machine steel. The results of these experiments are presented in Table 7, and are plotted in Figs. 14 and 15.

56 Experiments with the same cutter are joined by a line. In both figures the 34-tooth line and the 18-tooth line came superposed within experimental error. The 26-tooth line came roughly parallel but lower in both cases. This appeared to establish the equality of cutter efficiency in the case of the 34- and 18-tooth cutters with different tooth spacings when given equal chip volume. The distinct inferiority of the results with the 26-tooth cutter to the results of both the others suggested some defect in the cutter. It was found that the side clearance was almost nil.[1] This was remedied and the tests were repeated in machine steel with the result that the

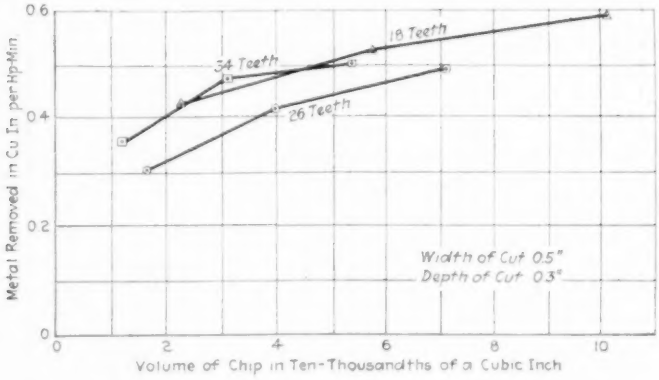


FIG. 14 EFFECT OF VARYING SPACING: ALLOY STEEL

26-tooth line coincided on the left with the two others. On the right there is a slight separation due to experimental error, but no consistency can be traced. The experiments with the 26-tooth cutter were not repeated in alloy steel, as that particular block was not available.

57 In a paper presented at the 1911 Spring Meeting of the Society by Mr. A. L. De Leeuw,[2] claims were made regarding the supe-

[1] Most of the tests in the present section consisted in cutting grooves 5 in. wide and 0.3 in. deep with a side mill. It was repeatedly shown that power consumption was not affected whether the cut happened to be a groove or a land, except in this one case of side-clearance trouble.

[2] Milling Cutters and Their Efficiency, Trans. Am. Soc. M. E., vol. 33, p. 245.

riority of wide spacing. This publication caused coarse-tooth cutters to increase in favor. If the experiments there recorded were to be compared on a chip-volume basis, the different cutters would doubtlessly lose their separate identity from an efficiency standpoint. As the speed of cutting is not recorded in these tests, it is not possible to find the chip volume, and so effect this comparison.

58 It is equally effective, however, to analyze data extracted from "A Treatise on Milling and Milling Machines," issued by the Cincinnati Milling Machine Company. On page 147 of that work results are given of tests with cutters of 22, 16 and 10 teeth, respectively. These results are there submitted as establishing

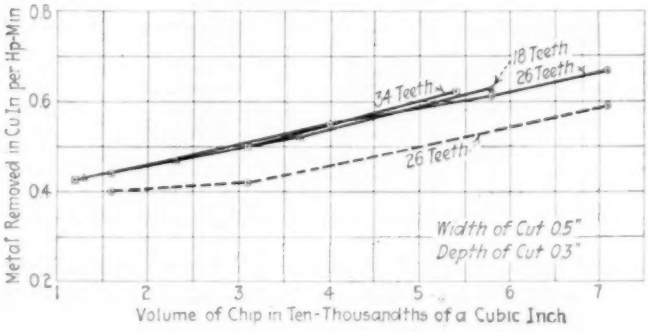


FIG. 15 EFFECT OF VARYING SPACING: MACHINE STEEL

conclusively the superiority of the coarse-tooth cutter. The data in question are reproduced in Table 8, the chip volume being calculated for 0.5 in. width of cut to make the results comparable with the others in this paper.

59 These results, shown graphically in Fig. 16, are seen to follow the same basic law as others already given. To summarize: Material and depth of cut remaining constant, the efficiency with which the metal is removed is determined solely by the chip volume and is independent of the number of teeth in the cutter.

60 If, in using a fine-tooth cutter, the feed be increased in order to give the same weight per chip as was given in a comparative test with a coarse-tooth cutter (and so equality of efficiency), one of three things might happen:

a Chips might clog due to lack of chip room
b The limitation of power of machine might be reached
c The limit of structural strength of the cutter might be reached.

61 Condition (*a*) is very apparent, for the efficiency rapidly drops to zero as soon as chip room becomes too small. This is proof that the spacing is inadequate, a condition which doubtlessly frequently existed formerly. Condition (*b*) is probably the indirect cause [1] of existing opinions regarding wide spacing of teeth. If in given tests, equal chips are taken, then the cutters will remove metal, the volumes of which per minute will be in the ratio of the

TABLE 8 CINCINNATI MILLING MACHINE COMPANY'S EXPERIMENTS

No. of teeth	Speed, r.p.m.	Feed	Volume of chip	Cu. in. per hp-min.
22	80.4	11.7	0.00062	0.442
22	80.0	14.8	0.00079	0.484
22	80.0	18.5	0.00099	0.577
16	82.0	11.9	0.00085	0.522
16	68.0	15.1	0.00130	0.631
16	67.0	18.6	0.00163	0.674
10	69.0	11.9	0.00162	0.625
10	67.0	15.1	0.00208	0.684
10	68.0	18.8	0.00259	0.731

numbers of teeth, and will of course call for correspondingly more power. Condition (*c*) would occur in the case of exceptionally thin cutters or of unusually deep cuts.

62 As many teeth as possible should be put into a cutter consistent with sufficient chip clearance. It is evident, of course, that sufficient excess chip space must be provided in a new cutter to allow for a reasonable amount of grinding. As soon as the necessary chip clearance is exceeded, possible economy is forfeited. A glance at existing high-powered cutters is sufficient to show that several times the necessary chip clearance is nearly always furnished.

63 The cuts taken in our experiments are throughout in an everyday production region. The Thrust and Torque Measurer was not capable of carrying spectacular cuts. However, that the same law holds in the region of heavier cuts is proved by the experiments plotted in Fig. 16, as they form a similar curve. Further, a study of these curves shows that chip congestion did not play any part with even the finest-toothed cutter, for this would have resulted in the last spot falling below the general trend of the combined curve.

64 Considering that the whole subject of milling action centers about the action of a single tooth, it appears desirable to think in

[1] See Par. 67 et seq.

terms of this. The authors take this opportunity of suggesting to milling-machine manufacturers that this point of view be encouraged in supplying instruction data with machines. As it is, the feed is invariably given in inches per minute.[1] This is useful in checking up the time required to do a given job after the feed is known. It is of no direct value, however, in checking up whether the feed is adequate for the work, because this is determined fundamentally by a definite feed per tooth.

65 Further, the feed per tooth is a function of the material and of the depth of cut. Data should be arranged so far as possible to focus attention on the maximum chip thickness. This is the basic criterion of effort from the cutter standpoint.

66 The authors believe that, in a few years, by coöperation between milling-machine and milling-cutter manufacturers, data

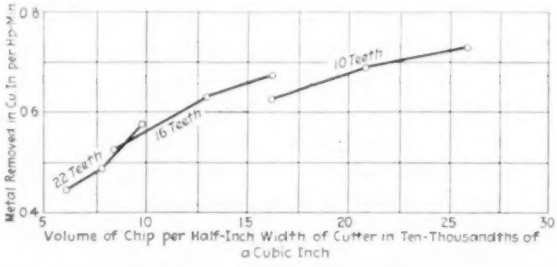


FIG. 16 CINCINNATI MILLING MACHINE COMPANY'S EXPERIMENTS

and practice will have evolved so that a set-up man will determine the appropriate chip thickness and then, by reference to tables, will set the speed and feed to give this. The feed per tooth for a given cutter determines the feed per revolution of spindle. If a chart were supplied giving the feed per revolution for all possible combinations of speeds and feeds, then this would be of direct use in

[1] The present practice is to give feed in inches per minute with machines having a constant-speed feedshaft. Feeds are given in inches per spindle revolution with machines having the feed train driven from spindle. Recent literature indicates that doubt exists regarding the relative merits of the two systems. The problem is purely and simply that of obtaining the greatest range with the simplest mechanism. The authors believe that the constant-speed feedshaft is unqualifiedly the better of the two systems. They further believe that this lends itself to a demonstration as rigorous as a proposition in Euclidean geometry does. For discussion of this, see letter to *Mechanical Engineering*, by John Airey, on Milling-Machine Feed Systems, March 1921, p. 207.

working from a feed-per-tooth starting point and would constitute the first step in the suggested direction.

67 *General Consideration of Removing Metal by Milling.* It is an interesting and profitable line of thought to consider, in its broadest phases, the problem of removing metal by milling. The first factor to be considered is chip thickness. In order to remove the maximum amount of metal per hp-min., the chips should be as thick as possible.

68 The chip weight is limited by the endurance properties of the cutting tool, as is also the cutting speed, which is the second factor to be considered. Various experimenters in lathe-tool action have shown that, for a given tool, the chip weight and the cutting speed are interrelated. It should be remarked here that the degree of bluntness permissible in a milling cutter is very slight, when compared with the allowable bluntness of a continuous-cutting tool.

69 The greatest metal-removing capacity occurs when the product of chip volume and cutting speed is at a maximum. Let us assume that the most advantageous combination of speed and thickness is known for a given tool and material, so that these factors may be kept constant. It then follows that the closer the teeth are together, the more metal will be removed per minute, provided, of course, that the spacing of the teeth is not so far reduced that clogging of the chips takes place. Since power efficiency is determined by chip thickness only, it is evident that this is not altered by varying the tooth pitch.

70 If it is possible to keep the feed per tooth constant and decrease the tooth spacing without exceeding the power of the milling machine, the ideal conditions are achieved, i.e., the maximum volume of metal is removed per minute with the expenditure of the least possible power. Suppose, however, that the limit of the machine's power is reached before the minimum possible spacing is obtained.

71 The problem now assumes a different aspect. We have already obtained the maximum power efficiency by virtue of using the thickest possible chip. If the tooth spacing is to be decreased, the power of the machine will be exceeded unless some of the other factors are changed. If the feed be reduced, the chip becomes thinner. This results in a decrease in the power efficiency coupled with a decrease in the amount of metal removed per minute.

72 Suppose that we leave the feed constant, decrease the tooth spacing, and at the same time reduce the spindle speed so that the feed per tooth remains the same. The power efficiency will

then remain constant, as will the amount of metal removed per minute. Furthermore, the decrease in peripheral speed will result in a lengthening of the life of the cutter between grinds.

73 We have thus demonstrated a priori that fine spacing is superior to coarse spacing, regardless of whether the cutting speed or the power of the machine be the limiting factor. In the first case, the fine-toothed cutter will remove more metal per minute. In the second case, the fine-toothed cutter will wear longer between grindings, due to the decrease in cutting speed.

SHAPE OF TOOTH

74 The following six features of a tooth will be discussed:

a Clearance	*d* Depth
b Rake	*e* Thickness at base
c Shape of back	*f* Clearance for chips.

These features are defined in Fig. 17.

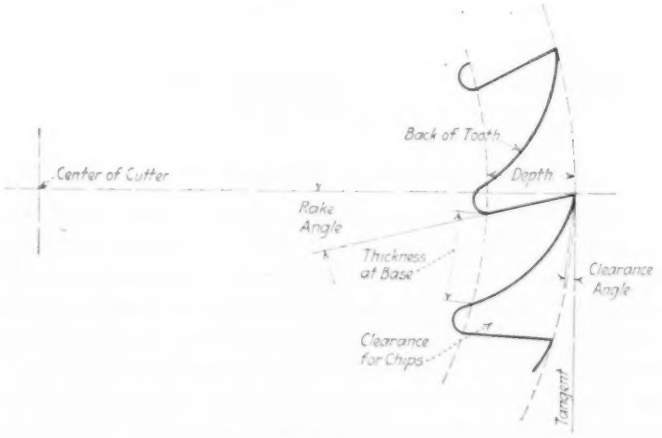


FIG. 17 DEFINITION OF TERMS

75 *Clearance.* Excessive clearance does not reduce the cutting force, does not appreciably affect the life of the tool in normal wear but *does* tend to chattering action; also it weakens the cross-section near the cutting edge and increases liability to snip. Clearance should therefore be kept to a minimum. Three degrees was found to be ample but 5 deg. was usually used. In many tool cribs it might be desirable to standardize at 8 deg., due to the possibility

of an error of a few degrees. In practice it is customary to speak of clearance in thousandths of an inch at the heel of a land. This is obviously a safer method, as the heel might bind for any given clearance angle if the land were sufficiently extended. However, for scientific analysis it is better to use degrees.

76 *Rake.* From a power-consumption standpoint, rake is increasingly beneficial as it becomes greater. The advantage of rake, though, does not increase at so fast a rate after about 15 deg. is passed. Further, this is influenced by the kind of material being cut. On the other hand, the life of the cutter might be influenced disadvantageously by excessive rake.

77 The action of wear may be classified, following F. W. Taylor's discussion on a lathe tool, into (1) simple abrasion and (2) influence of heat. It is believed that the greater the rake, the longer the life, from a simple abrasion standpoint. An increase

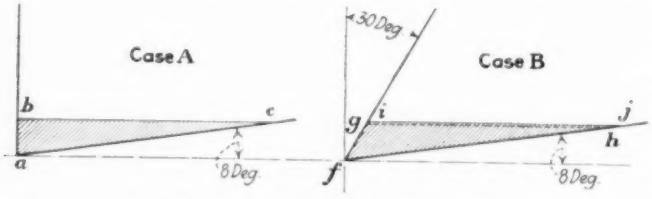


FIG. 18 INFLUENCE OF RAKE ON WEAR

in rake, however, necessitates a decrease in the heat conductivity of the tooth, as a consequence of which increased wear due to rising temperature is more likely to occur. A compromise can be obtained only by experiments on endurance and this is treated later.

78 In substantiation of the theory regarding simple abrasion action, consider the two examples of tooth, *A* and *B*, given in Fig.18. *A* has no rake and *B* has 30 deg. rake. Wear will take place parallel to the direction of cutting. The point will be eventually worn down to line *bc* in case *A* and to *gh* or *ij* in case *B*. It seems reasonable that the width of the abraded surface is a measure of the difficulty of forcing the tool into the work, and therefore of bluntness. Further, it seems reasonable that the volume of the tool abraded is a measure of the time required to bring the tool to that degree of bluntness.

79 Line *gh* is in line with *bc* and is shorter, on account of the rake in case *B*. Draw *ij* such that it is equal to *bc*, and therefore gives an equal degree of bluntness. The shaded area in case *B* is greater than in case *A* (because the two triangles have equal bases

but the triangle in B has the greater height). Therefore the tool B has a longer life. The triangle in B is 9 per cent larger than that in A.

80 The value is not excessive and is given merely as an explanation of why we do not find a decrease in endurance in the early stages of rake increase, as one might very easily expect from snap judgment.

81 Another and probably the most serious effect of excessive rake is the liability of the cutter to hog in, particularly in starting a heavy cut when there might be a resultant *upward* force exerted by the cutter. We can definitely ascertain the effect on power of

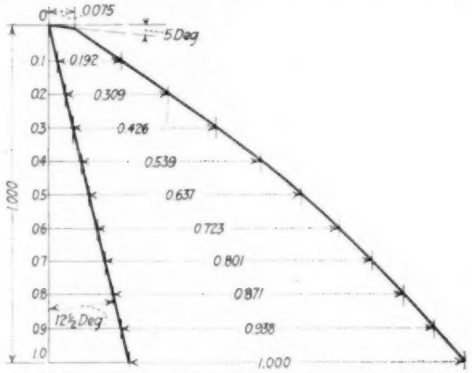


FIG. 19 IDEAL SHAPE OF TOOTH

a change in rake, but the other attributes are not so definitely ascertainable and must evolve protracted experience.

82 While the decided power advantage afforded by rake is indisputable, it is not advisable to jump too far in the higher regions. Had the importance of developing cutter design in this direction been acted upon decades ago, as it should have been, then we should be on a much safer footing, as a considerable amount of data would have been available regarding action of greater rakes in all types of situations. In Par. 81 the tendency to hog in was referred to. The Lincoln connecting rod (see Par. 109) presents a particularly vicious example of this tendency and was chosen for this reason.

83 Under the circumstances the authors believe and recommend that cutters be given $12\frac{1}{2}$ deg. rake; further, that for any definite job it might be very desirable to go beyond this, even as high as 25 deg., but that $12\frac{1}{2}$ deg. may safely be taken as a minimum, with the possible exception of rare and unusual jobs.

84 *Shape of Back.* The ideal tooth is that which is just as

likely to break in one place as in another. Any deviation from this shape is providing useless excess strength and is detracting from chip room or deleteriously increasing tooth spacing.

85 Analyzing the various tooth shapes that have been or may be used for the purpose, in the light of the engineers' theory of bending, we find that a tooth with a parabolic back furnishes uniform strength. If the tooth be radial and a concentrated force act normal to the cutting face, then the vertex of the parabola is at the cutting edge. As rake is increased or as the force is taken other than tangential, the vertex of the parabola becomes more remote.

86 An ideal tooth shape is given in Fig. 19, with construction data.

87 In Par. 85 the force was taken as concentrated. Actually the force is spread over a small area at the point of the tool. The assumption has been made that this area extends from the cutting edge to one-tenth of the depth. Consideration of this distributed force changes the curve over the one-tenth depth and further causes the parabola, if produced, to pass through the cutting-edge radius of one-twentieth the depth. A land is desirable to back up the cutting edge against shearing action. A straight line is drawn from the heel of the land tangential to the parabola. This also has a distinct advantage over the parabola alone, as it gives a longer grinding life before the land becomes excessive.

88 Due to the questionable legitimacy of the engineer's theory of bending when applied to short cantilevers (as in this case), conclusions derived therefrom can only be taken as a guide for experimental investigation. Twenty-four tools of high-speed steel were therefore shaped in accordance with above analysis and hardened. These were tested to destruction, and it was found that they broke with a sufficient variety of planes of weakness to justify the belief that the proposed shape has no serious place of consistent greatest weakness. In some cases the break was by snipping, in others in the body of the tool, again in others cleavage occurred roughly parallel to the back, and finally some broke in several places at once. The last is of course the ideal case.

89 *Depth.* The depth is controlled partly by the length of grinding life expected, but chiefly by the spacing, which in turn will be shown to depend on chip clearance. It is not possible by any method of calculation to predetermine the chip space necessary for given conditions of cut. To provide net cross-sectional area between teeth equal to the area of the heaviest cut would be absurdly in-

sufficient. The space actually needed is a function of the shape of the chip, depending on just how tightly it rolls, and manifestly a condition not lending itself to computation.

90 This feature can be determined only by experiment, but one fact should be kept clearly in mind in considering this, viz., that the *shape* of the space between teeth is equally important with its net area. The shape should be such as to prevent the chips from "packing," for this quickly causes the power to increase abnormally. Narrow, wedging bottoms must be avoided.

91 The prevention of packing is facilitated by a spiral cutting edge; also, packing is less liable to occur in thin cutters than thick ones. Thus, for a given diameter of cutter, narrow cutters, such as slitting saws, should have the most teeth, the number gradually decreasing as the thickness increases up to about 0.5 in. thick and then remaining constant. This is approximately true of present practice.

92 *Thickness at Base.* For a given depth, the thickness at the base is controlled by the ability of the tooth to resist fracture. This is, in turn, dependent on the quality of steel. Thickness at the base is taken as equal to depth in the recommendations here given. This is fairly consistent with present practice and gives an ample factor of safety for cuts that are customary.

93 *Clearance for Chips.* Chip clearance should be sufficient to accommodate the heaviest cut that it is desired to take and no more. It has previously been shown that if clearance is more than necessary or if the cutter is used for chips smaller than the maximum possible, then metal-removing possibilities are being sacrificed. Chip-clearance requirements govern the spacing, and this in turn fixes the number of teeth in a cutter for a given diameter.

94 *Design Formulas.* Formulas for determining the number of teeth for a known diameter of cutter and for determining the depth are given below, and a geometrical construction for obtaining shape of tooth has been given in Fig. 19. The arguments demonstrating the rationality of these are given in Appendix No. 2.

Let n = number of teeth
R = radius of cutter in inches
d = radial depth of straight face in inches.

Then
$$n = 19.5\ R^{\frac{1}{2}} - 5.8 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [1]$$
$$d = 0.215\ R^{\frac{1}{2}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [2]$$

An example is given in Fig. 20 of a 6-in. cutter.

95 The foregoing is for cutters 0.5 in. thick and upward. For cutters narrower than this, the number of teeth could be increased somewhat. Further, it is possible that if the spiral is excessive, say over 20 deg., the quicker clearing action would make more teeth possible.

A STATICAL STUDY OF SINGLE-TOOTH ACTION

96 Objections will doubtlessly be raised to the use of statical experiments, due to the supposed discrepancy between cutting action at a dead-slow speed, or in installments, and cutting action at a customary speed. The authors believe this discrepancy to be more imaginary than real and that considerable information can be obtained by statical studies which is not obtainable by other methods.

97 *Method of Experimenting.* The driving belt was removed, a pair of long clamps were bolted to the hub of the pulley and the

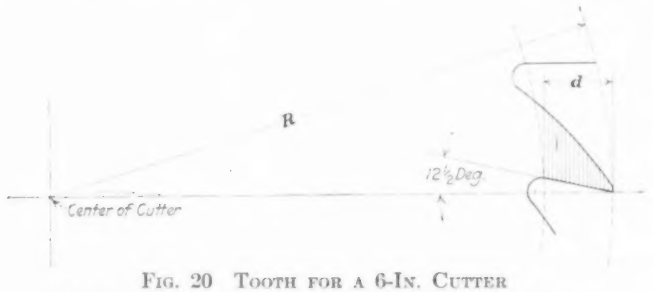


FIG. 20 TOOTH FOR A 6-IN. CUTTER

$R = 3$ in.; $n = 28$; $d = 0.372$ in. Shape of tooth taken from Fig. 19.

pulley was divided circumferentially into 50 equal divisions by center-pop marks and numbered; also a zero pointer was attached to a stationary part. The driving pulley could then be turned by hand. By means of the 50 subdivisions on the pulley and the gearing of the machine, the spindle could be turned through any desired small fraction of a revolution. The formation of a chip was divided into a convenient number of equal intervals and readings of the mercury columns were taken at each interval. The cut was therefore taken in a number of very small movements, the cutting stopping while a reading was taken. The results obtained were the same as would have been obtained by dead-slow continuous cutting. There was no observable change in the mercury columns when motion ceased.

98 The only object aimed at in this statical work, complete results of which are at hand, was to find if instantaneous chip thick-

ness is the sole criterion of force at a given instant. In the early part of this work the habit was formed of thinking in terms of chip volume. It was believed that this was the basic starting point for correlation of results; and so long as depth of cut was kept constant, there was no reason to question the legitimacy of this viewpoint.

99 As stated in Par. 43, the work done with various depths of cut was of doubtful value. Nevertheless, it was apparent that shallow cuts showed up more favorably on a chip-volume basis. This led to further speculation, ending with the conviction that not chip volume but maximum chip thickness was the feature of basic interest.

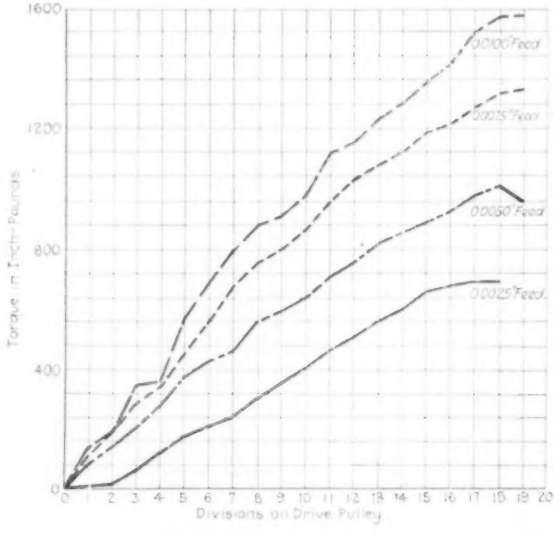


FIG. 21 STATICAL EXPERIMENTS ON FOUR DIFFERENT CHIP THICKNESSES

100 The truth of the above statement seems so obvious when the action of milling is carefully considered, that the fact would not be elaborated on, were it not that months of experimental work had been done before this idea developed. All chips, to within a small error of mathematical interest only, increase uniformly from zero thickness to a maximum. As any given instantaneous thickness of chip connotes a certain force-action at that instant, and as the increase in thickness is uniform, it follows that the maximum chip thickness is a unique criterion of the material removed per hp-min. for a given cutter and material.

101 On this assumption a miscellaneous set of experiments with constant material and constant shape of cutter tooth would fall on one curve, if metal removed per hp-min. were plotted against maximum thickness. The variation in depth of cut and the varying tooth spacing would have no influence.

102 *Comparison of Four Chips of Varying Feed.* A cutter of 10 deg. rake was used, the depth of cut taken was 0.3 in. and the

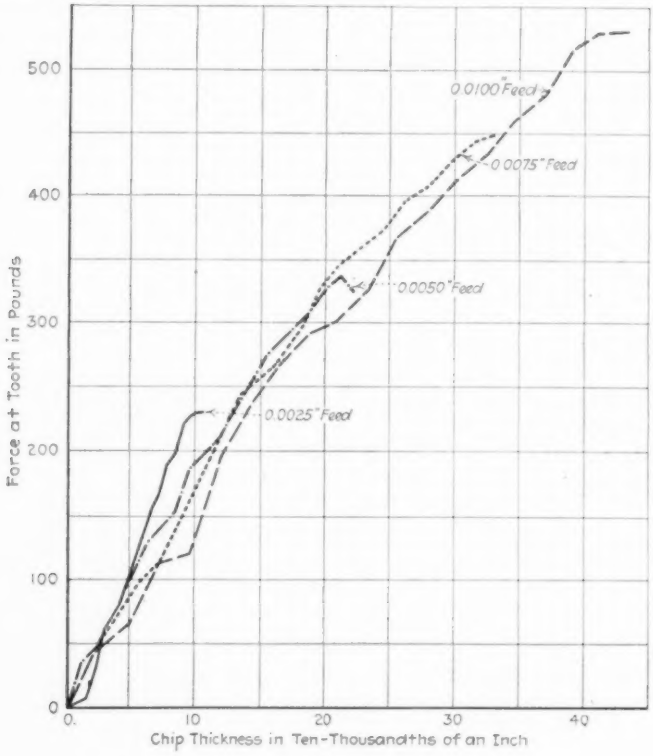


FIG. 22 FOUR STATICAL EXPERIMENTS SUPERPOSED

feed was taken successively as 0.0025 in., 0.005 in., 0.0075 in. and 0.010 in. The feed was obtained by carefully moving the table by hand and then locking the gib screw. The rotation of cutter necessary just to take one chip was divided into 20 equal parts and readings were taken.

103 The results are shown graphically in Fig. 21, the torque being plotted against divisions of the driving pulley. The same

general shape is observed except in the case of the 0.0025-in. chip. Several attempts were made to obtain consistent results with this without much success. A little reflection on the probable distortion of results produced by the knee bracket being, say, 0.0002 in. too high or too low, on a thin chip like this will account for this difficulty. These curves are rearranged in Fig. 22 to give Tangential Force plotted against Chip Thickness.

104 The finest chip again is somewhat irregular. Apart from this the chips are identical in force requirement for equal chip thicknesses. After all, it would be difficult to understand if a condition other than this were apparent.

105 Continuing the above point of view, it is seen that cutter efficiencies for two distinct jobs should be compared on the basis of equal maximum chip thickness instead of metal removed per minute. The latter is too crude and might give equality of performance when in reality the cutters are being far from equally punished. Further, to compare metal removed per hp-min. in two cases where thickness of chip is not identical, is grossly unjust and will lead to fallacious conclusions, e.g., the present superiority claimed for wide spacing.

ENDURANCE TESTS UNDER COMMERCIAL CONDITIONS

106 Although increasing the rake angle is unquestionably beneficial from a power standpoint, the same is by no means obviously true from the standpoint of endurance. In fact, it is apparent that a continuous increase of rake must ultimately jeopardize endurance.

107 To investigate this, numerous experiments were necessary and arrangements were made to make them under commercial conditions. Two jobs were experimented with, the Lincoln connecting rod and the Hudson gear-shift finger.

108 The only feature under consideration was the influence of rake. All other features were therefore kept constant. The speeds and feeds in regular use on both jobs were taken, but the cutters used were of different rake values. All cutters used were made at the same time from the same bar of steel and received the same heat treatment.

109 *Lincoln Connecting-Rod Job.* Fig. 23 shows this connecting rod, the part machined in these tests being indicated by a heavy line. The cutters used were $2\frac{5}{8}$ in. in diameter. This enables the cutter to form-mill the radius at the big end of the rod. The operation of the fixture is such that the work is fed vertically into the cutter until cutting is taking place over the full arc. When the

proper depth is obtained, the longitudinal feed is put in and the length traversed.

110 Starting in the rough over the full extent of the curved part is particularly severe on the cutter. This job is unusually well fitted for testing any tendency of a cutter to hog in. No trouble was encountered in this respect. The carbon content of the steel aver-

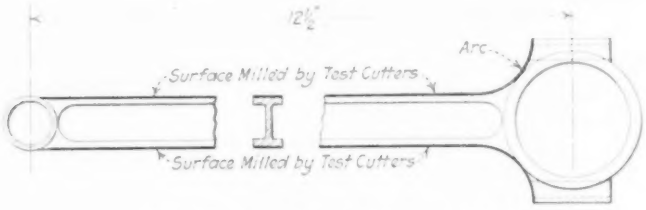


FIG. 23 LINCOLN CONNECTING ROD

aged about 0.50 per cent. The average depth of material removed was $\frac{3}{32}$ in., at a spindle speed of 125 r.p.m. and a feed of $6\frac{1}{8}$ in. per min. or 0.0041 in. per tooth.

111 In all, 901 rods were milled under observation. This furnished five grinds with the 0-deg. and 15-deg. rake cutters, eight grinds with the 10-deg. rake cutter, and two grinds with the 20-deg.



FIG. 24 HUDSON GEAR-SHIFT FINGER

rake cutter. The regular operator determined when the cutter was "blunt," this being judged by the roughness of the finish. The following results were obtained:

Rake angle, deg.	0	10	15	20
Average number of rods per grind	11.8	58.1	50.6	62.0

112 The 20-deg. cutter snipped in two teeth after the second grind. Ten to twenty times the number of experiments would be

required to state with any reliability the quantitative relation between rake and endurance. It is apparent, though, that for this material the influence of rake on endurance is at least beneficial up to between 10 deg. and 15 deg.

113 *Hudson Gear-Shift Finger.* The results of this job are unfortunately almost valueless. Due to a change in design, the part in question ceased to be manufactured after 2125 had been machined. The number machined by one cutter without grinding was unexpectedly large. This resulted in only one grind being possible with a 5-deg., 10-deg., and a 15-deg. rake cutter. Fig. 24

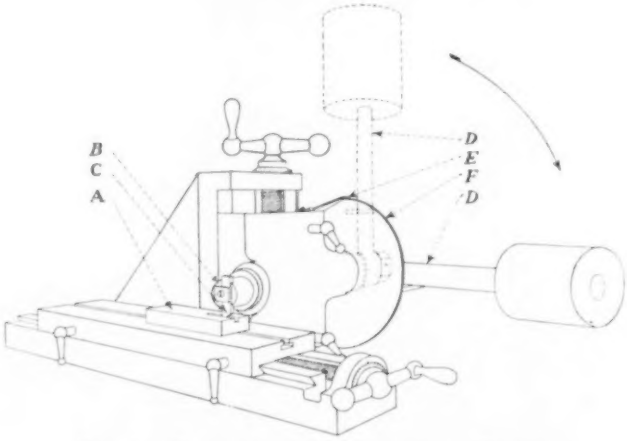


FIG. 25 DIAGRAM OF MACHINEABILITY TESTER

shows the finger with the part machined in heavy line. The results were as follows:

Rake angle, deg.	5	10	15
Number of fingers per grind	700	800	625

114 As a result of these experiments it may be said that from an endurance standpoint rake is certainly increasingly beneficial up to 10 deg. and probably is up to 15 deg. This statement is made definitely for steel up to 0.50 carbon, and it is believed that it will be found true for all metals.

EXPERIMENTS WITH MACHINEABILITY TESTER

115 The Machineability Tester was designed, as stated earlier, for measuring quickly the machineability of different metals. This machine is shown in Fig. 25.

116 In principle it is of course identical with the Chip Investigator, so far as measuring the integral effect of the action is concerned. Within its limitations, however, it is a much more efficient

TABLE 9 ENERGY AS A FUNCTION OF MAXIMUM CHIP THICKNESS

Material, cold-rolled steel; radius of cutter, 1¼ in.; width of cut, ½ in.

Depth of cut, in.	Feed per chip, in.	Maximum thickness, in.	Energy, ft-lb.	Cu. in. per hp-min.
0.04	0.004	0.000849	1.5	0.352
0.04	0.008	0.001697	2.4	0.440
0.04	0.012	0.002546	3.2	0.494
0.04	0.016	0.003395	4.0	0.528
0.08	0.004	0.001194	2.6	0.406
0.08	0.008	0.002389	4.4	0.480
0.08	0.012	0.003583	5.7	0.555
0.08	0.016	0.004778	7.0	0.604
0.12	0.004	0.001455	3.9	0.406
0.12	0.008	0.002910	6.0	0.529
0.12	0.012	0.004366	8.0	0.594
0.12	0.016	0.005821	10.0	0.634
0.16	0.004	0.001670	5.1	0.414
0.16	0.008	0.003338	7.8	0.541
0.16	0.012	0.005028	10.8	0.587
0.16	0.016	0.006677	13.5	0.626

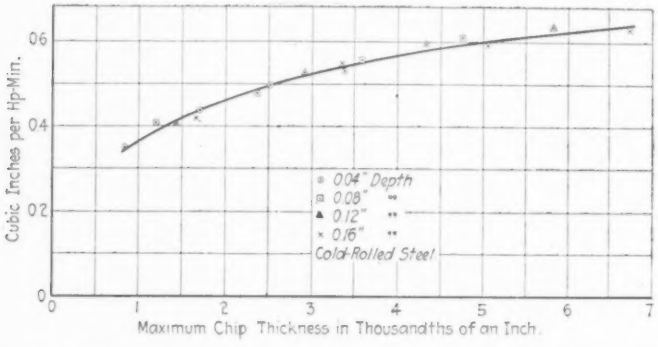


FIG. 26 METAL REMOVED AS A FUNCTION OF MAXIMUM CHIP THICKNESS

instrument than the Chip Investigator, as no calculation is necessary. If the weight be allowed to fall freely from the top position, it will rise almost to the same height. A pointer indicates the highest position to which the weight rises and this is taken as the zero reading. If, in a subsequent fall, a chip be formed then the weight will not rise as high. This change in height is a measure of the energy.

given up in forming the chip. The same principle is used in impact-testing machines.

117 One group of experiments run on this machine are recorded, as they were carried out to emphasize the analysis brought out previously that maximum chip thickness is the sole criterion of efficiency for a given tool and material. Four different depths of cut and four different feeds were taken, giving sixteen differently shaped chips in all.

118 The results of these sixteen experiments are given in Table 9 and Fig. 26, in which cubic inches per horsepower-minute are plotted against maximum chip thickness. It is seen that efficiency is a function of maximum chip thickness alone.

119 The importance of the above point of view cannot well be overemphasized. In arranging instructions for the operation of milling machines, in compiling speed, feed and depth-of-cut data, or in comparing different jobs, the fact should be kept in mind that maximum chip thickness is the true criterion of cutter-tooth punishment.

APPENDIX NO. 1

CHIP INVESTIGATOR — METHODS OF COMPUTING

120 The record shown in Fig. 27 is analyzed by means of a traveling microscope, using a foil templet to catch three points on a wave. There are two cases to consider: first, an analysis leading to a force-space curve; second, an analysis giving the energy absorbed, which of course is really the area under the first curve.

TABLE 10 COMPUTATION OF FORCE-SPACE CURVE

Experiment No. 32, on cast iron; feed, 0.016 in.

No.	Reading	Velocity	Acceleration	Dynamic force	Static force	Total force
1	172	403				
2	575	414	11	−355	372	17
3	989	418	4	−129	374	245
4	1407	420	2	−65	376	311
5	1827	420	0	0	378	378
6	2247	419	−1	32	379	411
7	2666	416	−3	97	380	477
8	3082	412	−4	129	380	509
9	3494	406	−6	194	379	573
10	3900	398	−8	258	378	636
11	4298	389	−9	291	376	667
12	4687	379	−10	323	374	697
13	5066	369	−10	323	372	695
14	5435	369	0	0	369	369
15	5804	381	12	−388	366	−22
16	6173					

121 The longitudinal position of every fifth wave was noted by means of a microscopic analyzer, which happened to read in centimeters. Referring to Table 10, we have:

Column 1, Serial number of point observed

Column 2, Position in tenths of a millimeter

Column 3, Successive differences

Column 4, Second differences; i.e., acceleration

Column 5, Dynamic force at the tool. This is equal to angular acceleration (reduced to proper units) multiplied by the moment of inertia of the oscillating mass

Column 6, Force required at tool to hold the arm statically

Column 7, Combined static and dynamic tangential force.

If column 7 be plotted against column 2, the units of the latter being suitably modified, a force-space curve is obtained.

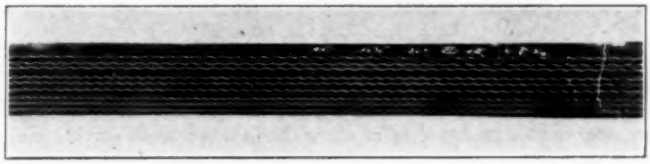


FIG. 27 EXAMPLE OF SMOKED-PAPER RECORD

122 *Energy Analysis.* This requires a knowledge of the angular velocity of the arm at the beginning and at the end of a cut but not at intermediate positions. These points on the smoked record are marked and the pitch of the

ENGINEERING SHOPS
University of Michigan

MILLING CUTTER RESEARCH

Experiment No. 451 Cut .003 In. Date 12-10 1920
Feed .0075 In. Spiral 31 Deg. Extra Weight 0 Lb.
Rake 10 Deg. Clearance 3 Deg

	A	B	C
Readings at Start of Cut	1 01	2 27	3 20
	0 00	0 00	0 00
Velocity at Start (V_1)	1 01	2 27	3 20
Readings at End of Cut	6 8 7 49	6 9 3 75	6 8 9 11
	6 8 4 55	6 9 0 19	6 8 4 85
Velocity at End (V_2)	2 94	3 56	4 26
$V_2^2-V_1^2$	7 6 24	7 5 05	7 9 08
Kinetic Energy given up in Ft-Lb.	1 97	1 95	2 04
Potential Energy given up in Ft-Lb.	3 54	3 54	3 54
Total Energy Used by Cut in Ft-Lb.	1 57	1 59	1 50

Record taken by E. E. O.
Record Computed by E.E.O.
Remarks: Cast Iron — ¼" Land

FIG. 28 COMPUTATION OF ENERGY PER CHIP

waves there measured. Five wave lengths are usually taken, straddling the point being investigated. These readings are then noted on a form, one of which is shown completely worked out in Fig. 28. In changing from $(V_2^2 - V_1^2)$ to kinetic energy given up in foot-pounds, a constant is involved which includes the moment of inertia and the change of units.

APPENDIX NO. 2

DESIGN OF CUTTERS

123 A semi-rational theory is here given for the design of cutter teeth. As a cutter increases in diameter, it is likely to take heavier cuts: a small-diameter cutter cannot be expected to carry the feed per tooth carried easily by a large slab mill.

124 If the reasonable assumption be granted that maximum chip thickness is proportional to the radius of the cutter, then the foundation is laid for a satisfactory theory of spacing.

125 Assume also that the force required to remove a chip is proportional to the square root of the thickness. This is equivalent to saying that a parabola can be fitted to the force-space curve of Fig. 3. Also

Let R = radius of cutter in inches
n = number of teeth
F = maximum tangential force
d = depth of tooth
t = thickness at base of tooth
f = maximum fiber stress permissible

126 The length of chip increases with an increase of radius, and as chip thickness has already been taken proportional to radius, the chip volume will vary as the radius raised to a power higher than unity. However, as the radius increases, the material becomes more compact for a given volume, due to the thickening of the chip, so chip clearance necessary may be taken as proportional to radius simply.

127 Since the shapes of the spaces between adjacent teeth are similar in cutters of different diameters, chip space is proportional to d^2, and as chip space is also proportional to radius as explained in Par. 126, it follows that:

$$d \propto \sqrt{R} \quad \ldots \quad [1]$$

also from Pars. 124 and 125,

$$F \propto \sqrt{\text{Chip Thickness}} \propto \sqrt{R} \quad \ldots \quad [2]$$

Therefore

$$d \propto F \quad \ldots \quad [3]$$

Assume for simplicity unit thickness at base of tooth. By the theory of bending,

$$Fd \propto t^2 f \quad \ldots \quad [4]$$

Therefore, by [3],

$$d \times d \propto t^2 f \quad \ldots \quad [5]$$

Or

$$\propto d \quad \ldots \quad [6]$$

Also, by the properties of similar shapes,

$$n = \frac{2\pi(R - d)}{t + Kt} \quad \ldots \quad [7]$$

where K is a constant.
Or, by [1] and [6],

$$n = \frac{2\pi(R - K_1 R^{\frac{1}{2}})}{K_2 R^{\frac{1}{2}}(I + K_1)} \quad \ldots \quad [8]$$

$$= K_3 R^{\frac{1}{2}} - K_4 \quad \ldots \quad [9]$$

where K_1, K_2, K_3 and K_4 are constants.

128 The constants K_3 and K_4 have been determined by observation of spacing in cases where chip room appeared adequate. This gives:

$$n = 19.5R^{\frac{1}{2}} - 5.8 \quad [10]$$

Also a relationship of

$$t = d \quad [11]$$

is simple and accords well with proportions repeatedly used and found amply strong.

129 Modification of K_5 in the relation,

$$t = K_5 d \quad [12]$$

affects the strength of the tooth in general. For instance, if a new superior steel were discovered, then K_5 could be decreased. Changing K_5 in no wise affects the rationality of Equation [10], but would indirectly influence the constants, leading to a greater n for a given R if K_5 be reduced.

Since, by [1], $d \propto R^{\frac{1}{2}}$, we find the following relation to give satisfactory results:

$$d = 0.215R^{\frac{1}{2}} \quad [13]$$

DISCUSSION

A. L. DeLeeuw. The highly interesting paper is of a dual nature. It is partly the record of experiments and observations which may be called its foundation, and partly a superstructure of comment and conclusions based on these observations. The writer makes this distinction because he is entirely in harmony with the aims and objects of the first part, while at the same time he wishes to state his objections to many of the conclusions reached in the second part.

The method of measuring the forces employed in the separation of a milling chip is very interesting and sheds a great deal of light on a hitherto obscure subject. It is to be regretted that the authors did not go a step further and attempt the separation of a chip by a very slow process, so as to enable them to observe the deformation of the metal at the different stages of separation, either by direct observation or by the moving-picture method.

The authors' determination of the direction, location and magnitude of forces in horizontal and vertical directions is somewhat along the line of Nicholson's experiments and, in the writer's opinion, deserves the same criticism. One of the chief requirements of machine-tool construction is rigidity, because it is well recognized that the lack of this rigidity increases certain of the forces or at least increases their effect on the work to an extent altogether out of proportion to the amount of lack of rigidity. Therefore, to place the work on movable pistons is to go away

from the very essence of machine-tool design. It may be countered that the method used is the only one available for the determination of the forces, but against this argument the writer would say that the magnitude of the forces found in this manner is probably not the magnitude of the forces which are at work in a well constructed machine tool.

The method of determining machineability is interesting as such, but it is not quite clear to the writer of this discussion what such a test can bring out. It would seem that this test merely indicates the amount of energy required to take a chip of predetermined size of the metal to be investigated, but this amount of energy has already been determined by the first test. Besides, in order to be of any value whatever, the nature of the tool must be standardized. It has been the writer's experience that very small differences in the surface of the tool, in its sharpness, and in the manner in which it is ground, have far-reaching effects on the ease with which a tool will separate a chip. The authors of the paper did not state whether they had any means of standardizing the nature of the tool.

Even if such a standardized tool could be made and maintained, it would be more than doubtful whether the tests would show the true machineability of the metal to be investigated. It is somewhat difficult to argue on this subject because machineability is not a term which can be expressed in simpler terms, although most of us know by experience what is meant by it. There are certain bronzes which would show a high machineability by the tests mentioned, but which would be very difficult to machine on account of the tendency of the chip to tear out of the metal. In other words, the metal is easily *cut* but it is not easy to make a *proper surface*. There are kinds of steel which do not require a great deal of power to cut when the tool is new, but which almost at once begin to build up a ridge of fine particles near the tool edge, so that in a very short time the tool no longer cuts but punches the metal off. Such materials are exceedingly hard to machine, but relatively easy to cut with a new tool. The machineability of wrought iron is very high if we look at the power required for cutting, but it is very low when we try to make a good screw thread. Many examples of this kind could be mentioned, showing that what the average man in the shop understands by machineability is not the kind of thing whch was investigated by the machineability tester of the authors of this paper.

Attention should be called to Fig. 3 which shows the relation between force and chip thickness. The line of chip thickness ignores the fact that the chip does not start in with a thickness zero, or that it winds up with a thickness less than maximum. The second objection is of minor importance, but the first objection relates to one of the main difficulties met in practical milling. A milling chip tends to start with a zero thickness and, consequently, the cutter must slide over the work for some distance before it will be able to penetrate. During that time the arbor or the cutter itself is made to spring away from the work until finally the amount of spring is sufficient to overcome the resistance of the metal against the cutter tooth. At that moment the actual cutting begins. As an example, a 6-in. cutter taking a feed of 0.010 in. per tooth must slide over the work for a length of 0.088 in. before the cutter is sprung away 0.001 in. Whether it requires a spring of 0.001 or 0.0005 in., or maybe 0.002 in. depends on a variety of conditions, and mainly on the sharpness of the cutter. We do know that when we have milled a piece of work with a plain or spiral cutter, we could take a finishing cut by merely moving the work once more under the cutter and that without making any adjustments. Other conditions which affect this amount of spring are spirality of cutter and depth of cut. If the cut is quite deep and more than one tooth is in the cut at all times, then the downward pressure caused by the heavy chip may completely counterbalance the upward pressure against another tooth trying to enter the work. Spirality affects it to the extent that a spiral cutter is all the time more or less in the same condition as a cutter of which more than one tooth is embedded in the work.

Another point to be considered in this relation is that the tooth enters the work suddenly, so that, as a matter of fact, the chip as it is actually produced does not start with a thickness zero. Instead of having the theoretical shape as shown in Fig. 29, it will actually have the shape as cross-hatched in Fig. 30.

The observations about rake are closely in accord with the results the writer found in tests made between 1907 and 1914. The existing differences may be explained away by the fact that in the writer's experiments part of the power consumed was used for feed.

The angle of 12.5 degrees as a compromise angle seems to be too large for gritty materials. The writer has been compromising on 10 degrees. However, there is room left for discussion on this subject

Referring to Par. 29, Effect of Clearance: Previous tests have shown the same results brought out by the authors, namely, that clearance has no effect on the power consumption. However, it does have an effect on the life of the cutter and the finish produced. It seems that gritty materials require more clearance than those materials which will produce a continuous chip. The reason is not clear, nor is there any law known by which the shop man should be guided as to the amount of clearance to furnish. Though 3 degrees would give a very good finish on steel, it does not seem

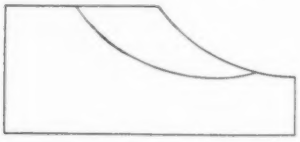


FIG. 29.

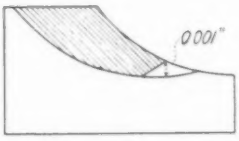


FIG. 30.

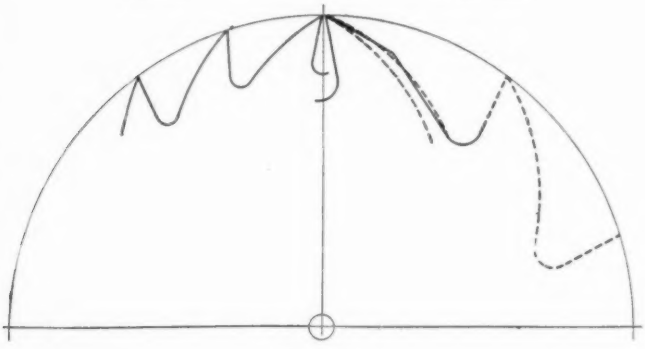


FIG. 31.

FIG. 29 THEORETICAL SHAPE OF CHIP; FIG. 30 ACTUAL SHAPE OF CHIP; FIG. 31 RECOMMENDED PROPORTIONS OF TEETH

to be practical to use this small angle on account of the great effect of a small amount of wear. Five degrees is sufficient for all practical purposes when steel only must be cut; 10 degrees is preferable for cast iron only; while 7 or 8 degrees seems to give good satisfaction as a compromise angle for cutters which must cut iron as well as steel.

In Par. 33 the authors state that as a chip thickens, metal is removed more easily per unit volume. As a practical rule, this statement is acceptable and meets all shop conditions. As a statement of accurate fact, it may be doubted. A number of observations made by the writer of this discussion showed that the

efficiency increases when the cross-section of the chip approaches more and more a perfect square. In other words, a chip $\frac{1}{8}$ in. x $\frac{1}{8}$ in. requires less power than one which is $\frac{1}{4}$ in. high and $\frac{1}{16}$ in. thick, and also less than another chip which might be $\frac{1}{16}$ deep and with a $\frac{1}{4}$ in. feed. The fact that the feed never exceeds the depth of cut except with finishing cuts makes the difference between the observation of the authors of the paper and the writer of this discussion of academical importance only.

Referring to Par. 34, the writer wishes to call attention to the fact that in a series of tests made by him in 1907 and 1908 the method of comparing input of power with output of chips was employed but that, at the same time, the mechanical efficiency of the machine was determined for various amounts of input and that for drive and feed separately as well as in combination. Though under such conditions we do not get as clean-cut a picture of the laws governing the removal of one single chip, we do get a better picture of the entire action of the milling machine and the laws governing the actual practice of milling. This remark is made with full appreciation of the merits of the investigation described in the paper under discussion. The writer is in full sympathy with investigations which aim to get at the root of the problem.

Referring to Par. 48: It must be acknowledged that there is no reason why the total amount of power required to remove a single chip by a spiral tooth of a cutter should be less than that required for a straight tooth. The conclusion reached by the authors of the paper might be accepted as practically correct for a narrow cutter with one single tooth. However, when the single tooth has considerable width, we meet a different problem. Following the action of such a single tooth we would find that it starts with one point of the cutter striking the work, refusing to penetrate and therefore sliding over the work. When sufficient pressure has been accumulated, the tooth enters and new points begin to slide. The first point which had entered gradually meets thicker and thicker material and the downward pressure increases until a point is reached where it balances the upward pressure. If the depth of cut is great enough and the chip thick enough, there may be a time when the cutter is actually forced downward. If there is only one tooth, this process will be repeated every time a revolution has been completed, but if there are more teeth, then the second tooth may begin to act before the first one is entirely out of the cut; and if this is the case the period of

sliding over the work may be materially reduced. Under such conditions, then, the waste of power due to this sliding is reduced and the efficiency of the cutter increased. In addition, a smoother cut will be taken and less metal will be left, due to spring. The main reason why the writer has advocated the 25-degree angle is that such an angle permits the simultaneous action of more than one tooth on relatively narrow work or with relatively shallow cuts. Even greater angles might be used but for the fact that end pressure begins to play a considerable part with the higher angles, and the further fact that one end of the tooth becomes too weak.

The writer must take issue with the statement contained in Par. 50 and following paragraphs. If the only point for consideration were the power required to remove a cubic inch of metal, the conclusions reached by the authors of the paper would be quite correct. However, in practice this power consumption is only one of the elements to be considered and not necessarily the most important one. In order to bring out the various elements which should be considered it may be best to take a concrete example.

We will suppose that a cut $\frac{1}{4}$ in. deep must be taken on a piece of mild steel 4 in. wide, and that we use for this purpose a cutter $3\frac{1}{2}$ in. in diameter, having 10 teeth. We will give a feed of 0.010 in. per tooth and use a speed of 66 ft. per minute, which requires 72 r.p.m. Under those conditions the feed per minute is 7.2 in. and the amount of metal removed per minute is 7.2 cu. in. This will require a machine driven by a 10-hp. motor.

If we should follow the recommendation of the authors and put more teeth in the cutter, let us say twenty, but otherwise use the same feed and speeds, we would require a 20-hp. motor. However, as the machine on which we wish to do the cutting is only supplied with a 10-hp. motor, we will be compelled to reduce the feed per tooth, and as this reduced feed per tooth makes the efficiency lower, we will have to overload the motor considerably in order to remove the original 7.2 cu. in. of metal. Or, if we do not wish to overload the motor, we will have to be satisfied with a reduced cut. However, the authors say that this difficulty might be readily overcome by lowering the cutting speed. If we should reduce the speed to 33 ft. per minute, we would again be able to give a feed of 0.010 in. per tooth and yet remove only 7.2 cu. in. of metal; and we would be working with the original efficiency.

At this point we must introduce a new element and that is the appearance or finish of the work. In a large portion of all milling

work, one cut only is taken which is required to produce a finish of sufficient fineness. This finish depends on the number of revolution marks per inch. Under the original conditions, that is with the 10-tooth cutter, these marks were 0.100 in. apart; but with the increased number of teeth and reduced speed they will be 0.200 in. apart, which would not be permissible under most circumstances. As a matter of fact, in a large percentage of all cases, the feed per revolution is limited by the desired finish at least as much as by the ability of cutter or machine to take the cut. The authors of the paper seem to be under the impression that the underlying idea for the introduction of the wide-spaced cutter was to supply sufficient chip space. As a matter of fact, this consideration only explains their origin. It was the observation of a case of insufficient chip space which induced the writer to try a somewhat wider spacing. Subsequent tests with these wider spaced cutters brought out further advantages.

In selecting the proper kind of cutter for a milling job, the following line of reasoning may be considered typical for a majority of cases:

The nature of the work compels us to limit the distance between revolution marks to, let us say, 0.050 in. In order to do the job as quickly as possible we must have as much feed per minute as possible. The feed per minute depends on two elements: cutting speed, and size of cutter. With the material to be cut and considering the depth of cut, we will be limited to a cutting speed of, let us say, 70 ft. per min. We must now select the smallest possible cutter which will do the work in order to get the greatest possible number of revolutions which will give 70 ft. cutting speed. Now enter such considerations as size of arbor, depth of teeth, size of keyway, all of which may prevent us from making the cutter as small as we would like to have it. If, for instance, we are compelled to use a fairly long arbor, we must make that arbor of sufficient size to avoid excessive bending and torsion. Having considered all these items we finally decide on the size of cutter. As we want to do the job with as little power consumption as possible, we must provide for taking the heaviest possible chips, and this in its turn means the smallest possible number of teeth in the cutter. It will be noticed that this line of reasoning is almost diametrically opposed to that followed by the authors of the paper. There are certain limiting conditions which prevent us from making the number of teeth as small as might be desir-

able. To have too great an angle between two adjacent teeth would mean that in practically no kind of cut would we have more than one tooth buried in the metal. To avoid the resulting hammer blow, we must put a sufficient number of teeth in the cutter.

The original reason for the reduction of the number of teeth of a cutter was chip space. Soon afterward it was found that the power consumption was smaller and that, as a result, a given-size milling machine was capable of taking heavier roughing cuts with the wide-spaced cutter than with the older type. Other advantages showed up later on. Among these advantages are the fact that under many conditions a wide-spaced cutter will finish more pieces before it becomes necessary to resharpen it. This is especially so when the teeth are made with rake, but is true also, though to lesser extent, when the teeth are formed without rake.

Not only does a wide-spaced cutter require less sharpenings, but it will stand many more sharpenings, so that the life of the cutter is very much increased and the cutter expense very much reduced.

Furthermore, it requires less time to sharpen a cutter with few teeth than one with many; and not only that, but it requires less attention and care on the part of the grinder hand in setting the wheel and cutter to the proper position. A great number of teeth in a cutter causes interferences between the grinding wheel and the teeth to be ground.

There is another element of economy in taking heavy chips beside the economy of power. To bring this out let us exaggerate actual conditions. Let us imagine a cutter working with a feed of 0.060 in. per revolution, and let this cutter have 60 teeth. Then each tooth takes a feed of 0.001 in. If now the sum of all the little imperfections which necessarily exist is as little as 0.001 in. some of the teeth of this cutter will take no cut and others may be called upon to take a cut of 0.002 in. Not only does this produce a very uneven action and a tendency to chatter, but the cutter will dull as if it were taking a cut of 0.002 in. because it must be considered dull when one tooth is dull. If this same cut had been taken with a cutter with six teeth, each tooth would have taken 0.010 in., and if the irregularities again amounted to 0.001 in., we may find that some of the teeth take 0.009 and others 0.011 in. per tooth, which is not very much faster than its proper rate of dulling.

It will be seen, then, that chip space and saving of power are only two of the elements to be considered, whereas the other ele-

ments mentioned are of at least as much importance in practical shop operations. The writer has gone somewhat at length into this matter on account of the one-sidedness of the exposition of the merits of wide-spaced cutter contained in the paper. It should be understood, however, that the writer does not recommend the widest possible spacing for all imaginable cases of milling.

Besides the reasons mentioned, there are other reasons which may prevent us from taking the heaviest possible chips. In many cases it is not possible to clamp the work down in a sufficiently rigid manner to withstand the pressures accompanying heavy chips. In other cases, the piece of work may be of such a character as not to allow heavy cuts to be taken, either because it is too frail or because the heat generated will distort the piece, or both. In order to obtain the greatest possible economy in the milling operation, we must give the cutter the greatest possible number of revolutions and, after we have selected the smallest practical cutter, there is only one source of economy left and that is to increase the cutting speed. It was considerations like these that led up to the system of stream lubrication whereby unusually high speeds can be obtained with some intelligence and care. As stream lubrication not only prevents the cutter from heating unduly, but also keeps the work cool, it has a double effect on the economy of milling operations.

The foregoing considerations lead the writer to take exception to the statements contained in Par. 62.

Fig. 31 shows the proportions of teeth as recommended by the authors and also those recommended for wide-spaced cutters. The sketch shows a $3\frac{1}{2}$ in. cutter. The left-hand side shows the teeth as formed by the formula given in the paper and with twenty teeth in the cutter. The right-hand side shows the teeth, in black, as they are actually made; and in broken line as they would be made according to the formula given in the paper. It will be seen that there is but little difference between these two shapes — the actual shape being slightly the stronger. The main difference, however, lies in the fact the broken-line shape requires a specially made cutter for generating the backs of the teeth, whereas the black shape can be formed by any cutter. When the land, as originally furnished, becomes too wide, it is a simple matter to grind the backs of the teeth with a cup wheel and to repeat this action as often as is necessary. So long as the size of the land is not reduced beyond the original size, the strength

of the tooth will not be impaired. It is this practical reason which makes the writer advocate the shape as shown on the right-hand side of the sketch.

R. Poliakoff. For measuring the torque and thrust, the

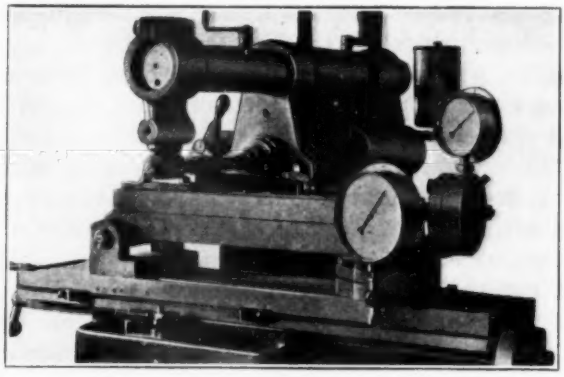


Fig. 32 Poliakoff Milling Machine Dynamometer, 1913

authors employ three pipes attached to the machine and filled with mercury. The pipes have a height of at least 72 in., and it can be readily seen that even under laboratory conditions the mercury

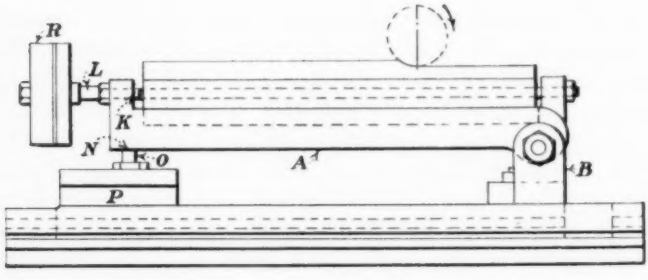


Fig. 33 Construction of Poliakoff Milling Machine Dynamometer

columns would jump up and down, first because of taking the cut, and second, because of the chattering of the machine. On account of this, the conclusions based on the readings of the columns must be to a certain extent doubtful.

As a matter of fact the method employed by the authors was used by the writer as far back as 1913, with this difference, however, that in order to eliminate the drawbacks of the mercury columns just pointed out, pressure gages were employed. Fig. 32 shows the apparatus of the writer mounted on a milling machine (which in this case happens to be a Cincinnati No. 2), and the principles of its construction are shown in Fig. 33.

The piece to be milled is bolted to a slide, the longitudinal movement of which is guided by grooves in the lever *A*, fulcrumed on points of screws passing through the lugs *B*, secured to the table of the machine. The slide bears against the point *K* of screw *L*, pressing the hydraulic support connected with a pressure gage. In a similar way the lever *A* bears on the point *N* of the screw *O* which transfers the pressure to the support *P* connected with another pressure gage.

When the cutter rotates in the direction indicated by the arrow, the force exerted by the teeth resolves into a longitudinal pressure acting parallel to the table and a downward pressure acting at a right angle to the table. These respective pressures are transmitted to the gages and therefore can accurately be determined, the degree of accuracy depending exclusively upon the size (diameter) of the gage dials.

The authors employ three pistons (or hydraulic supports) instead of two as employed by the writer. The purpose of the third piston (hydraulic support) is to take care of changes in the pressure when the table travels from left to right. The writer accomplished the same object by setting up an initial pressure in one of the vertical pressure gages, thus eliminating the other (third) one, and making the whole arrangement simpler.

Of course, as in the case of the authors, the leverage of the downward pressure in relation to the fulcrum, about which the lever swings, changes as the table moves. But, as the feed is known, the leverage of this pressure can be determined at any moment; in other words, the writer's methods to read this were identical with the authors'. However, the writer recognized early that such a method of constant observations and calculations is far from perfect, because the arrangement itself is applicable only in case of milling cutters having their teeth parallel to the axis of the cutter, when there is no third component of the cutting force acting alongside the arbor parallel to the table, as is the case, for instance, with milling cutters with spirals.

In order to have the readings of the gage correspond to the downward pressure independent of the location of the cutter itself, the principle of the platform scale was resorted to by the writer. As everybody knows, when an object is weighed on a platform scale the weight registered is always the same, no matter on what part of the platform the object is placed. Figs. 34, 35 and 36 show the latest dynamometer of the author built on this principle, for which he holds a United States patent acquired from him by The Cincinnati Milling Machine Company, Cincinnati, O. It consists essentially of a work platen supported at each end by a wide Emery plate fulcrum (Fig. 34), the lower end of which rests on two levers, which carry a definite portion of the vertical load on the platen to a hydraulic chamber placed centrally under the work

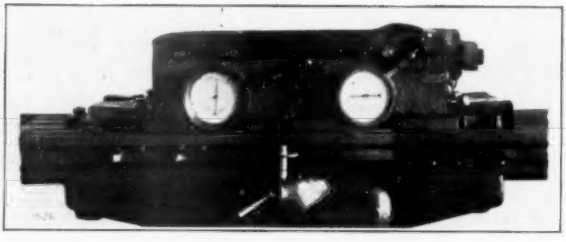


FIG. 34 LATEST TYPE POLIAKOFF MILLING MACHINE DYNAMOMETER

table. The chamber is connected with the left-hand gage which is graduated by trial in terms of the vertical load in pounds. The horizontal load is transmitted through bars which are flexible vertically to the crosshead seen at the right in Fig. 35. This crosshead transmits the load to a hydraulic chamber between the crosshead and the end of the main frame of the dynamometer, the hydraulic chamber being connected to the right-hand dial gage. Heavy springs place initial loads in each chamber, so that they will show loads in either direction.

The Emery plate fulcrums carrying the loads to the levers are so constructed as to be rigid against vertical and cross loads but flexible to longitudinal loads and the bars to the crossheads are flexible to vertical loads so that neither system interferes with the action of the other. Guards are provided so that any desired lubrication or flooding of the cutter may be used.

When it is desired to obtain the transverse pressure, that is,

the pressure in line with the arbor of the machine, the dynamometer is mounted on the table as shown in Fig. 35 and the pressure ascertained again from the right-hand dial. This arrangement enables the end thrusts of spiral and face-milling cutters to be determined.

The dynamometer shown mounted on a Cincinnati No. 5 heavy milling machine has the capacity to withstand longitudinal loads of 25,000 lb. in one direction and 4,000 lb. in the opposite direction, vertical downward pressures of 10,000 lb. and upward



FIG. 35 DYNAMOMETER MOUNTED FOR MEASURING TRANSVERSE AND VERTICAL PRESSURES

pressures of 7,000 lb. The readings are exact. The whole arrangement is very sensitive. Whatever the pressure, there is no wide fluctuation and disturbance of the dial hands, and therefore the readings can be very easily and accurately taken.

As to the remarks of Mr. DeLeeuw that the power consumed by the cutter is not the only factor that has to be considered in milling: this is, of course, true and in determining the cost of a milling operation, one will find that the item of power consumption constitutes but a small percentage of the total. However, even a small power consumption may be the result of a great cutting force and such a force affects both machine and the cutter very materi-

ally. Hence the necessity, or, at least, the advisability of studying these forces, for which study the dynamometer of the writer, just described, offers ready means. It can be installed on any milling machine, hence the possibility to use it both in the laboratory and the shop, especially so because a laboratory study of milling operations must, of course, have in view the practical shop end as the final goal.

There is another field of investigations for the dynamometer of that kind in the present day when standardization is aimed at so much. Certain materials can be established as standards, if

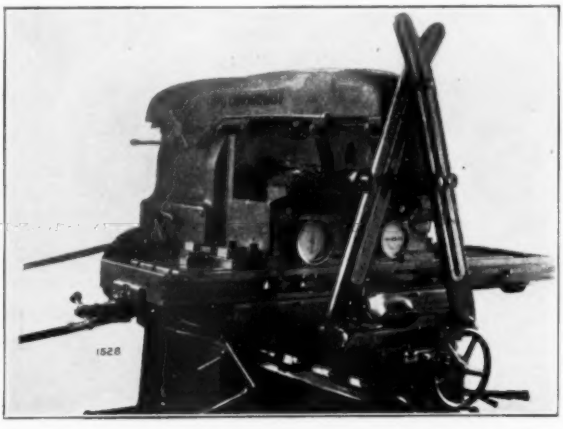


FIG. 36 DYNAMOMETER MOUNTED FOR MEASURING HORIZONTAL AND VERTICAL PRESSURES

under certain prescribed conditions of speed, depth of cut, feed and cutter, they will register certain readings on the dials of the device. It will be seen that with this possibility, there is no need for the machineability tester of the authors, since all the objects of the latter are more readily met with by the dynamometer.

EARLE BUCKINGHAM. The authors are to be congratulated on the valuable contribution they have made to our knowledge of milling. Two or three points have struck the writer very forcibly and he will therefore confine his discussion to these points.

He wishes to add his voice to the authors' in their plea that our engineering schools investigate shop processes as well as

problems of design. Judging from his own experience, the average graduate starts his practical work under the impression that shop processes are a finished subject, and that but little fundamental improvement is possible. It is only after several years of experience, particularly if investigations are made as to why certain conventional shop practices are followed, that realization of the fact that we know practically nothing about the fundamental principles of our basic shop practices is obtained. Investigations to determine the fundamentals of any of our shop practices, such as the authors of this paper have made, offer a practically virgin field of investigation, and one which the writer has found by his own experience in a few minor researches to be intensely interesting as well as to offer many chances of valuable practical results.

The writer is very much interested in the results obtained when using a spiral cutter. He ventures to offer two possible reasons why these should show less effective results. One possible reason might be on account of the negative rake on one side of each cutter. The writer is under the impression that these tests were made by cutting grooves in various materials.

Again, it is possible that the power required for milling depends also upon the length of the cutting edge in action. A spiral fluted cutter has a longer cutting edge than a straight fluted one. Possibly both the longer cutting edge and the negative rake on one side of the cutter are responsible for its lower efficiency as regards the power consumed.

If the length of the cutting edge is a factor in regard to the efficiency of milling, this might explain in part some of the results obtained when using a cutter with a higher rake. The increased rake unquestionably increases the efficiency. If the increased length of cutting edge on the sides due to this increased rake reduces the efficiency, it would explain in part some of the results. Take for example the tests on bronze in Par. 20. If these tests consisted in cutting grooves three-tenths of an inch deep, we would have the following lengths of cutting edges on the sides:

0-deg. rake, length of sides, 0.600 in.
10-deg. rake, length of sides, 0.609 in.
20-deg. rake, length of sides, 0.639 in.
30-deg. rake, length of sides, 0.693 in.

The lengths of the faces are equal. If we consider the cutting action on the sides where no rake exists as equivalent to the cutting

action of a zero-degree rake cutter, it might explain in part the drop in the efficiency of the highest rake cutter.

The authors state, however, in a footnote to Par. 56 that it was repeatedly shown that the power consumption was not affected whether the cut happened to be in a groove or on a land. This seems to eliminate the foregoing explanation. In this event, the explanation may be that the area of the surface of the cutter in contact with the chips becomes greater as the rake angle is increased thus creating more friction. This might explain the results obtained from using lubrication. The chips from some materials curl up more than those from other materials. If the chip clung closely to the face of the cutting edge, the lubricant might cause the chip to wring to the face in much the same manner as two gage blocks are wrung together, thus greatly increasing the friction. An analysis of the character and shape of the chips produced from different materials with the use of cutters of varying rake angles might shed some light on this. At all events, the subject is one which could be investigated further to great advantage and it is to be hoped that the authors of this paper will see their way clear to carry their excellent work further.

There is just one more point which the writer wishes to discuss. In Par. 73, referring to the advantages of fine spacing for the teeth of cutters, the authors state that "the fine-tooth cutter will wear longer between grindings, due to the decrease in cutting speed." Too much emphasis cannot be given to the fact that a decreased cutting speed greatly increases the life of a cutter between grindings. The proportional increase in life is much greater than the proportional increase in speed. The writer is acquainted with some very interesting tests made some five years ago to determine the effect of cutting speed on the life of a cutter. Tests were made at speeds of 70 and 80 feet per minute with high-speed steel cutters on small nickel-steel forgings. The results were very consistent and showed that when running at 70 feet the cutter would mill over twice as many pieces as when running at 80 feet. Thus two cutters of the same diameter, taking the same feed per tooth so that each would remove the same amount of material in the same time, etc., but with 21 teeth in one, for example, and 24 teeth in the second, would show a great difference in their life between grindings. If the conditions were identical to those mentioned above, the 24-tooth cutter would mill twice as many pieces. This should make apparent the importance of tooth-numbers in milling-cutter design.

CARL G. BARTH. The writer has barely found time to study the paper in a very superficial manner, but as the subject it treats has been an object of his own attention ever since he became substantially interested in the art of cutting metals, he feels it incumbent upon him to discuss it to some extent. This he can best do by briefly reviewing his own connections with practical milling problems during the past twenty years.

Ever since the writer's direct collaborations with Mr. Fred W. Taylor in the field of metal cutting and the construction of the slide rules first described in the paper,[1] Slide Rules for the Machine Shop as a Part of the Taylor System, these particular slide rules, which are directly applicable to all machines that use round nose roughing tools, viz., lathes, boring mills, planers, shapers and slotters, have been of inestimable value to him in increasing production on this class of machines.

It must have been in about the year 1904 that the writer realized that there was no essential difference in the action of a lathe slicing or cutting-off tool and each tooth of a milling or gear cutter, and that any consideration of what a milling cutter ought to do would have to be based on what each tooth ought to do. This once clearly realized, the writer inside of a week, constructed a slide rule for pre-determining suitable feeds per revolution of cutters of different shapes and taking different depths of cut; basing this on the strength of the teeth and allowing a factor of safety to be determined by numerous trials; and a second rule for the determination of the proper speed to run a cutter with the feed thus determined, by applying with suitable modifications for the contour of the teeth, the laws governing a lathe cutting-off tool.

The studies made at that time necessarily impressed the writer with the advantages of a suitable per-revolution feed series as against a per-minute feed series, in running milling and gear-cutting machines on a scientific basis, which necessarily begins with a consideration of feed per tooth and the number of teeth in the cutter.

But the writer has only recently attempted to design a milling slide rule that also takes the power into account, and it will be shown how a formula for the power required to take a milling chip in steel was developed, by falling back on experiments made about 1882 by Messrs. Fred W. Taylor and Wilfred Lewis jointly

[1] Trans. Vol. 25, 1904, p. 49.

on the pressure exerted on a lathe cutting-off tool working under different feeds. The formula is as follows:

$$P = 81{,}000\ w\ t^{0.7} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [1]$$

This formula, the writer feels certain, contains fully as much of the truth required for practical purposes as anything that can be obtained by experiments of the kind made by Messrs. Airey and Oxford, whose main conclusions, at any rate, differ in no way from conclusions long since drawn from power experiments with lathe tools, the general behavior of which can in no way differ essentially from that of the teeth in a milling cutter.

The experiments just referred to are given below just as they were copied in 1899 from Mr. Taylor's personal records, together with formula [1] worked up at that time to represent them approximately, and a comparison of the tool pressures figured by that formula and the actual pressures obtained in the experiments.

TABLE 11 EXPERIMENTS ON A TURNTABLE PLATE OF STEEL $2\frac{7}{8}$ IN. THICK

Width of Cut, Inches	Feed, in.	Vertical Pressure on Tool	
		Actual	By Formula [1]
$2\frac{9}{16}$	0.005	5053	5087
$2\frac{9}{16}$	0.01	7852	8263
$2\frac{9}{16}$	0.01	8192	8263
$2\frac{9}{16}$	0.01	8231	8263
$2\frac{9}{16}$	0.015	10,222	10,975
$2\frac{9}{16}$	0.015	9996	10,975
$2\frac{9}{16}$	0.02	13,486	13,434

Tensile strength 85,787 lb., elongation 11 per cent. carbon, 0.485, rake angle of tool (back slope) = 10 deg., clearance angle = 3 deg.

Referring now to Fig. 37, in which f = feed per tooth of a milling cutter, t = approximate theoretical thickness of shaving at the distance $R - x$ from the center of the cutter, h = total depth of cut, w = width of rectangular groove cut; R = radius, D = diameter and N = number of teeth, in cutter we have

$$t = f \sin \alpha$$

and accordingly, by formula [1] above, the tangential pressure at this point of the chip

$$P_\alpha = 81{,}000\, w(f \sin \alpha)^{0.7}$$

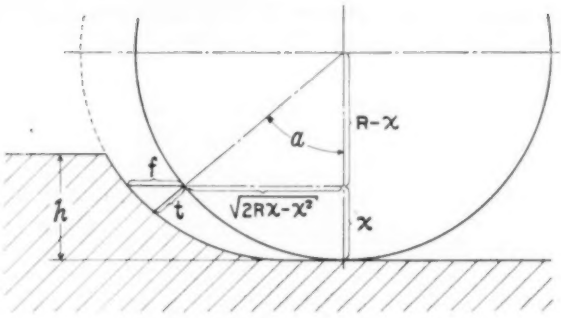


FIG. 37. ANALYSIS OF CUTTER

From this we get the differential of the work of removing the whole chip to be

$$\delta W_a = P_a \cdot R \cdot \delta a = 81{,}000\, w \cdot f^{0.7} R \cdot \sin^{0.7} a \cdot \delta a,$$

$$W_a = 81{,}000\, w \cdot f^{0.7} R \cdot \int_{a=0}^{a=a} \sin^{0.7} a \cdot \delta a. \quad . \quad . \quad . \quad . \quad . \quad [2]$$

Unable to solve the integral exactly, the writer solved it approximately by plotting the curve representing $\sin^{0.7} a$ and calculating, by the repeated use of Simpson's rule, the area (integral) between the limits $a = 0$ and $a = \frac{\pi}{18}$, $a = 2\frac{\pi}{18}$, $a = 3\frac{\pi}{18}$, and so on up to and including $a = 9\frac{\pi}{18} = \frac{\pi}{2}(90°)$.

Then plotting these areas as ordinates to their respective superior limits of a as absissas, these formed a curve whose true equation was found to approximate the simple expression [1] $\sqrt{\frac{a^3}{\pi}}$

[1] COMPARISON OF VALUES FOR $f \sin^{0.7} a\, \delta a$.

a		Value of $f \sin^{0.7} a\, \delta a$	
Degrees	Radians	As Calculated by Simpson's Rule	As Expressed by $\sqrt{\frac{a^3}{\pi}}$
10	$\pi/18$	0.03002	0.04114
20	$\pi/9$	0.09746	0.11636
30	$\pi/6$	0.19277	0.21376
40	$2/9\pi$	0.31088	0.32910
50	$5/18\pi$	0.44765	0.45994
60	$\pi/3$	0.59928	0.60460
70	$7/18\pi$	0.76204	0.76189
80	$4/9\pi$	0.93224	0.93085
90	$\pi/2$	1.1062	1.1107

which therefore is also an approximation to the $f \sin^{0.7} \cdot \alpha \cdot \delta\alpha$.

Substituting this in Eq. [**2**], we then have approximately

$$W_a = 81{,}000\, f^{0.7} \cdot w \cdot R \int_{\alpha=o}^{\alpha=a} \sqrt{\frac{\alpha^3}{\pi}} \quad \ldots\ldots\ldots \quad [3]$$

But $\alpha = \sin^{-1} \frac{\sqrt{2Rx - x^2}}{R}$, a very close approximation to which is [1]

$$\alpha = \pi \sqrt{\frac{x}{5R - x}}$$

This substituted in the expression $\sqrt{\frac{\alpha^3}{\pi}}$ gives

$$\sqrt{\frac{\alpha^3}{\pi}} = \sqrt{\frac{\pi^3}{\pi}\left(\frac{x}{5R - x}\right)^{\frac{3}{2}}} = \pi\left(\frac{x}{5R - x}\right)^{\frac{3}{4}},$$

which value, finally substituted in Eq. [**3**] together with the full depth of the cut h for x, and $\frac{D}{2}$ for R, modifies this to

$$W_h = 40{,}500\, w \cdot f^{0.7} D\pi \cdot \left(\frac{h}{5R - h}\right)^{\frac{3}{4}} \quad \ldots\ldots \quad [4]$$

This then is an expression for the work done by each tooth of the milling cutter of diameter D, cutting the straight groove of width w and depth h, with a feed per tooth f.

[1] This is derived from the following remarkably close algebraic approximation to cos α in terms of α in radians, which the writer arrived at more than twenty years ago, but which has never before been published.

$$\cos \alpha = \frac{1 - 4\left(\frac{\alpha}{\pi}\right)^2}{1 + \left(\frac{\alpha}{\pi}\right)^2}.$$

It gives exact values for $\alpha = 0$, $\alpha = \frac{\pi}{3}$ (60°), $\alpha = \frac{\pi}{2}$ (90°). Between $\alpha = 0$ and $\alpha = \frac{\pi}{3}$, the greatest error is less than 0.18 of one per cent, namely too small; while between $\alpha = \frac{\pi}{3}$ and $\alpha = \frac{\pi}{2}$ the approximation is too great in an increasing ratio, which, however, only approximates the limit $\frac{16}{15\pi} = 1.186$ as both the approximation by the formula and the true value of cos α approach 0 as a limit when α approaches $\frac{\pi}{2}$ as a limit.

If the number of teeth is N, the feed per revolution will be $F = f \cdot N$, or $f = \frac{F}{N}$

The total work done by all the teeth in a revolution can now be written

$$W = 40{,}500\, w \cdot \left(\frac{F}{N}\right)^{0.7} D\pi \left(\frac{h}{5R - h}\right)^{\frac{3}{4}} \cdot N$$

$$= 40{,}500\, wF^{0.7}N^{0.3}\left(\frac{h}{5R - h}\right)^{\frac{3}{4}} \cdot D\pi \quad . \quad . \quad . \quad . \quad . \quad . \quad [5]$$

and hence the average circumferential resistance to the rotation of the cutter is

$$P_c = 40{,}500\, wF^{0.7}N^{0.7}\left(\frac{h}{5R - h}\right)^{\frac{3}{4}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [6]$$

This formula forms the basis for the power portion of the writer's latest and complete milling slide rule for use in steel.

FRANK B. GILBRETH. Of the papers presented and of the remarks of the discussors, the writer has one criticism to make.

All of the conclusions are based on what one thinks after he has figured out as best he can the causes of the observed results There is a field open to the makers of machine tools that has not been utilized in the least which enables one to know the causes of these results and which substitutes information for speculation.

The photographic record of the tool and the chip furnishes the one best method of obtaining such information. With the new and specially devised apparatus now available it is possible to take stereoscopic records at high speed with a penetrating screen which record the chatter and loose motions and wearing and working of the machine, and the jarring of the building and the bending of the tools and the behavior of the chip. This process is not patented. It is available and ready for use.

You are only guessing and drifting when you are not acquiring and utilizing these fundamental data.

JOHN AIREY. Mr. DeLeeuw raises various points in his exceedingly valuable contribution to the discussion. Regarding the absence of dead slow experiments, reference to Par. 96 shows a section entitled, "A Statical Study of Single-Tooth Action." In this work we aimed primarily at a dead slow cutting action. Preliminary experimenting showed that the column readings did for convenience a series of statical studies as this was easier than

taking continuous readings. We believe the observations would be identical with the observations of dead slow experiments. However, a detailed study of chip deformation was not made and this we believe would be well worth while.

The objection raised to mounting the work on moving pistons is purely a quantitative consideration and not a qualitative one. To refer to them as moving pistons is erroneous as the movement but slightly exceeds the ordinary elastic movements of a machine. Such movement as might occur is explained purely by compressibility of water, because the movement of liquid in the observation column is negligible; this being throttled by a damping action. Furthermore, the ratio of the cross sectional areas of the piston and column respectively is more than 10,000 to 1. Lack of rigidity, if it exists to a sufficient extent, makes its presence felt by causing chatter. When chatter occurred in an experiment, conditions were so changed as to remove this or the experiment was rejected. Further, attention is drawn to the fact that an initial force of any required degree can be used by means of the tension rod and spring, shown in Fig. 11. The claim that magnitude of force will be changed is not well founded. The matter of rigidity in all cutting operations is one of degree only and the absence of it can be measured by chatter.

The criticism made of proposed tests for machineability is well founded in stating that the proposed test compares only energy requirements, but does not measure truthfully the difficulty of producing a satisfactory surface. To this extent the proposed test ignores a feature which is very desirable. Nevertheless, few will dispute that if we are able truthfully to measure the machineability of metals from the standpoint of energy only, (restricting ourselves to roughing operations) a very large territory is covered; in fact, by far the most important, as was repeatedly experienced in the roughing of shells. As to the difficulty of maintaining a standard tool, this difficulty does not exist. For purposes of comparison, any geometrically clearly defined and perfectly sharp tool would suffice. The same doubts as expressed by Mr. DeLeeuw were experienced by the experimenters in the early stages. It was found, however, that the energy required by a tool to take a certain size chip was not affected by a gradual loss of initial not change on stopping the motion completely: hence, we took sharpness for more than a hundred chips, so that it was of no practical moment.

Regarding the cut not actually starting where diagrammatically we should expect it to do so; this is doubtlessly true. In view of this fact, the force-space curve of chip formation, shown in Fig. 3 was expected to show a hump in its early stages, because of the resistance of the material to the cutting edge getting under the skin. The impossibility of finding this effect leads one to conclude that the frictional resistance during the sliding period is not greater than the resistance to cutting would have been.

The point raised that clearances must be greater for gritty materials is well taken, although in these particular experiments experience was not had in gritty materials.

The authors disagree with Mr. DeLeeuw in stating that the efficiency is the greatest when the chip is a perfect square. Throughout the early part of the experimenting outlined in this paper, a constant depth of cut was maintained. The practice was

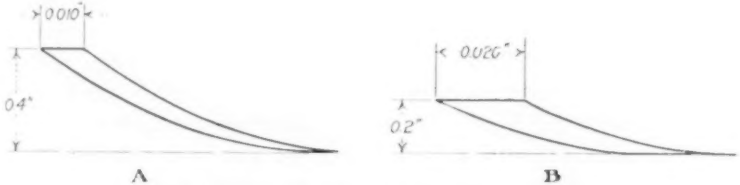


FIG. 38 COMPARISON OF DEPTHS OF CUTS

adopted of plotting all data on a chip volume basis. All results followed substantially smooth curves. Later it was seen fit to run some experiments with a different depth of cut. The results of these experiments, when plotted, showed that they did not belong to the same family as the previous ones but agreed amongst themselves. This led to the thought that chip volume was not the desirable quantity to take as the independent variable, because it was apparent that the energy consumed was not a single valued function of this quantity. Fig. 38 shows two chips of the same cross sectional area, but of different feeds and depths. The energies needed in the two cases are quite different. The explanation is not difficult as we have already emphasized the fact that thick chips are more economical than thin chips and the two sections shown differ widely in average thickness of chip. This lead to the idea that if maximum chip thickness be taken as the independent variable, then all results would fit one curve for the same material and per inch width of cut. To substantiate this

point of view 16 experiments were run under different conditions as shown in Fig. 26 of the paper. These results, we consider, prove conclusively that maximum chip thickness is a unique criterion for energy required to remove 1 cu. in.

In reference to the suggestion that spiral cutters having two teeth in action at once might account for the increase in efficiency, this would be acceptable if it had been possible to show that the sliding of a tooth, prior to entering, absorbed more energy than if it had entered the material. But this was not found to be the case.

In reference to Mr. DeLeeuw's interesting discussion on the advantages of wide spacing, the authors take issue with his reasoning. Perhaps this would be stated more accurately by saying that his reasoning is accepted but the premise taken is disputed. The premise in question is that the distance of revolution marks is the all important factor, rather than the metal that can be removed per horsepower-minute. This premise is clearly not applicable at all to roughing cuts. That it is applicable to many cases of finishing cuts, is not disputed. There are, however, many cases of finishing where this consideration does not completely control. Further, in considering finishing cuts, the power consumption is a very minor consideration and this field was not considered by the authors.

The statement is made that wide-spread cutters require fewer sharpenings, other conditions being equal. An explanation of this appears difficult. In comparing two cutters with different numbers of teeth, then if the feed per minute remains constant, a single tooth in each cutter will take the same number of chips but the coarse tooth will take a heavier chip. It seems scarcely reasonable that the heavy chip should dull the cutter less than the lighter chip (excluding, of course, such things as sand action). If feeds are adjusted so that the two cutters take the same feed per chip and also remove the same cubic inches per minute, then it is indisputable that the cutter with the greater number of teeth will dull the more slowly because each tooth takes fewer chips per minute, and the chip weights are identical.

The discussion showing the supposed disadvantages of closer spacing from the standpoint of irregularities in grinding is again hinged entirely on the premise of a fixed maximum feed per revolution and this the authors do not accept.

In the figure given by Mr. DeLeeuw, comparing the different tooth shapes, it is not stated why the solid line is supposed to be a

stronger shape than the dotted line. The strength of a chain is the strength of the weakest link; so surely any deviation from uniformity of strength is undesirable.

The objection to the use of a curved cutter is not a good commercial argument. Actually it has proven that the manufacturing costs are cheaper because with the special cutter for a curved back tooth only one cut is taken whereas with the two flats, two cuts must be taken in manufacturing cutters. The initial cost of special cutters is serious only in the small job shop and the shape of tooth as recommended is naturally intended only for the quantity production of milling cutters.

Mr. Poliakoff refers to the vibration of the mercury column. This vibration could be adjusted to any required degree by means of a damping device. For certain studies the unrestricted mercury movement was a distinct benefit because it indicated the irregularities in the cutter. For continuous reading this vibration was damped out, thus getting an average value.

Mr. Poliakoff's reference to his instrument developed for measuring end thrust and downward pressure is rather unfortunate, in that it clouds the issue. Mr. Poliakoff's machine is a very valuable standardized commercial article. The thrust and torque measurer of the authors is solely a research apparatus. The difference in action is considerable. It is just as impossible to obtain power with this type of dynamometer, by means of only two observed quantities, as it is impossible to solve for three unknowns by means of only two equations. Mr. Poliakoff's dynamometer will not measure power. If, however, *assumptions are made regarding the location* of the resultant force, then power could be computed from the observed data. The measurement of this location was one of the authors' chief troubles. If this location be assumed, then the investigation could not be classed as *research*, excepting the object be the study of feeding pressures longitudinally.

Regarding Mr. Poliakoff's suggestion of using his instrument as a machineability tester the same comments made above hold good.

The authors wish to express their appreciation of Mr. Buckingham's remarks regarding the lack of fundamental research on manufacturing methods. The theory advanced regarding the loss of power with spiral cutters cannot apply to any of the experiments with the chip investigator as these were invariably lands and not grooves.

CARL J. OXFORD. There is but little that the writer can add to what Professor Airey has already said, as he has covered all main points of our paper.

The writer should like, however, to call attention to the results of additional research work which has been carried on since this paper was written.

The authors began to consider the question of the operating economy of milling cutters from the standpoint of power consumed, and found that this power factor, which is seldom considered in connection with a milling operation, is really a very important item as far as actual dollars and cents are concerned.

The reason why the question of power required for a milling cutter has received so little attention in the past is perhaps that power is a sort of intangible quantity in the average shop. It is usually charged up to overhead without bothering to look into details as to how it is used. However, it is found that the saving of power, even on a single milling operation, is an item not to be disregarded.

A milling cutter requiring 15 hp. consumes in a nine-hour day:

$$9 \times 15 = 135 \text{ hp-hr.}$$
$$135 \times \frac{746}{1000} = 100 \text{ kw-hr.}$$

This power consumption, by the way, is not at all uncommon for a single production milling operation.

Supposing that it is possible so to design this cutter that 20 per cent in the power consumed can be saved, then 20 kw-hr. a day will have been saved.

Figuring power at 3 cents per kw-hr., as is done in Detroit, the saving for a single cutter is 60 cents per day.

Certainly this is an attractive figure, especially in shops where hundreds of milling operations are performed every day. In production milling, where one type of cutter is used continuously, it is therefore distinctly worth while to study both design and operation, in order to effect the most economical use of power.

No. 1805

STRESSES AND DEFORMATION IN FLAT CIRCULAR CYLINDER HEADS

A MATHEMATICAL ANALYSIS

By Gilbert Dudley Fish,[1] New York, N. Y.
Non-Member

An analysis covering homogeneous elastic disks, where form and loading are symmetrical with respect to all diameters, where the loading is a combination of fluid pressures and of forces acting normally on concentric circles, where the thickness is uniform, and where all strains are within limits of true elasticity. Formulas applicable to all cases considered in the paper are developed, and equations are given for the constants of integration involved in the mathematical analysis. Two typical cases of the application of the principles involved have been analyzed and the results presented diagramatically. To facilitate the computation of critical stresses and deflections in cases commonly occurring, simplified formulas are presented which may be solved by the use of charts from which certain terms are derived.

THE author made this analysis while employed by the Nitrate Division of the War Department, during the winter of 1920–1921. The work of which this investigation formed a part, was the redesign of the synthetic ammonia plant known as U. S. Nitrate Plant No. 1.

2 The special conditions causing most of the mechanical difficulties of this process, are the operating pressure of 100 atmospheres and the operating temperature of 500 deg. cent.

3 It has not been found necessary to make any special analysis for cylinder walls, to accompany this analysis of cylinder heads, because the standard treatment for thick cylinders applies.

4 It is desired to acknowledge the valuable suggestions and other assistance of Capt. R. S. Tour, chief of Plant No. 1 Section, and Capt. A. E. Hecker, Mechanical Engineer in charge of design.

INTRODUCTION

5 This paper is a mathematical analysis of stress and strain, and contains no discussion of the merits of the physical laws on

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Presented at the Annual Meeting, December 1921, of The American Society of Mechanical Engineers.

which it is based. It adopts the conception of internal stress as a dependent function of strain, quite distinct from internal force which follows the laws of equilibrium; it considers that internal stress, as distinguished from internal force, is the criterion for resistance.

6 The field in which this analysis is intended to be useful is the design of large cylinder heads to withstand high pressures. Boiler practice, dealing with light plates and moderate pressures, provides no formulas or other guides which may be used properly in cases of pressures as high as one ton to the square inch, or of heads one foot or more in thickness. The futility of applying empiric formulas or other approximations of unknown latitude, in the design of a cylinder head which is to cost thousands of dollars, and the failure of which might cause immense damage, becomes apparent when it is discovered that in many familiar cases the values of critical stress, as given by various commonly employed methods or formulas, run as low as one-half or as high as double the values obtained by rigorous analysis.

7 A feature of this analysis which is thought to be of special importance is the treatment of a cylinder head having a central hole. Such a hole, even if small, has a tremendous effect on the critical stress.

8 While the study of the elastic deformation of a head is necessary preliminary to the determination of stresses, it is not for this reason alone that the analysis of deformation is important; the effectiveness of a gasket, depending as it does on residual gasket load, is directly related to the deflections of the head at the gasket circle before and after application of the working pressure.

9 It is thought proper to answer in advance an objection which may be raised against the formulas resulting from this analysis. In case it be attempted to check some of the stress formulas, by using them to calculate the maximum intensity of tensile or compressive stress in a plate or cylinder head which is known to have failed under a certain loading, a highly erroneous result is to be expected. This analysis is confined to the conditions existing in a body which is not strained beyond the limit of true elasticity at any point. The physical laws connecting strains and forces are not even approximately true outside the limits of elasticity, and the internal stresses, unlike those in a simple beam, are dependent on deformation. Therefore this analysis does not yield formulas whereby the ultimate resistance of a disk may be predicted. It

is important that the strains in any cylinder head should not exceed the limits of elasticity, and it is consequently not essential to know what loading would cause it to crack or burst. In design, there must be established limiting intensities for stresses of tension, compression and shear, all well within the respective limits of elasticity for the material at whatever temperatures may be expected.

10 While it is not within the scope of this paper to make recommendations in regard to practice, it is thought proper to discourage the use of the ultimate strength of a material (whether in respect to tension, compression or shear) as a criterion for the maximum permissible stress in practice. Either the elastic limit or the fatigue limit is preferable as a criterion, especially in case of a special alloy steel or other material for which there exists no standard practice in regard to working stresses.

11 Attention is invited to the relatively large effect of detrusive (shearing) strain on the deformation of a thick head. Whereas, in the case of a boiler head, the deflections due to detrusive strain are minute in comparison with those due to flexure, they may even exceed those due to flexure in the case of a cylinder head designed to withstand several tons to the square inch, especially if there be a hole through the head.

12 Reference is made to the mathematical treatment of flat circular plates, by Prof. F. Grashof of Carlsruhe, published in his Elastizität und Festigkeit. There is believed to be no flaw in his analysis, and the present study, originally made without reference to his work and from a somewhat different method of approach, gives results identical with his, in so far as the same ground is covered by both. However, modern industrial apparatus involves conditions quite different in degree, if not in principle, from those contemplated by Professor Grashof, so that his formulas cannot, without a great deal of mathematical work, be extended so as to apply to actual cylinder heads. For example, it is necessary to analyze shear deflection, which was of no importance when pressures did not exceed a few hundred pounds to the square inch. Again, in the case of high pressure, surface application of load, as contrasted with distribution of load throughout the mass, appreciably reduces compressive stresses and deformation, and should be considered. The only special cases for which Professor Grashof published resulting formulas (i.e., formulas which can be used by direct substitution of numerical data), are very elementary cases which are never even approximately realized in the type of apparatus here considered.

It will be found, therefore, that only a small part of this analysis is concerned with ground already covered in the work referred to. If part or all of the remainder duplicates anything previously published, the author can only plead ignorance.

PRELIMINARY EXPLANATION

13 Hooke's law, and Poisson's principle of constant ratio between lateral and longitudinal strains, are used as bases for this analysis. It is further presupposed that the elastic properties of the material are the same at all points and in all directions; also that elements (lines of particles), which in the absence of strain are straight and normal to the neutral surface, do not suffer curvature or obliquity due to flexure within the limits of elasticity, but do become curved and oblique to the neutral surface due to detrusion alone.

14 To define the general case to which this analysis is applicable, the initial presumptions are set down as follows:

a The head is of uniform thickness throughout

b The head is simply and uniformly supported around a circle.

The familiar case of fastening by means of bolts, closely and uniformly spaced around a bolt circle, there being clearance between head and flange, is nearly a perfect realization of this assumption, except in respect to the unimportant direct stresses near the bolt circle; the shearing stresses at the bolt-circle section, which may be critical, are closely approximated by the formulas if the bolt-hole area be deducted from the gross area of the section. The design of a cylinder head may sometimes suggest fixation of the head at the support, as in the case of a head bolted tightly against a flange; however, in high-pressure work such fixation cannot actually exist, for no bolts are so inextensible as to prevent the slight angular strain which destroys fixedness, although a small degree of restraint may be imposed. Therefore fixed heads are ruled out of consideration as non-existent in fact, although it is not difficult to analyze the stresses and strains which would exist in such heads. Partially fixed heads are also excluded, for one reason because it is not possible to estimate the degree of fixedness in any given case, and for the other reason because such manner of support is considered bad practice, there being no known means for combining it with a gasket or other device so as to make a joint tight against high pressure.

c The head may be solid (i.e., have no perforations other than

bolt holes); but, if not solid, it has only a single central hole of uniform circular bore, concentric with the bolt circle. If there be such a hole, no matter how it may be plugged, stuffed or capped, the head is considered unrestrained around the hole, for the same reasons for which it is considered simply supported around the margin, on the ground that no pipe, rod, cap or other device can have more than a slight restraining effect. Any slight degree of fixedness which may actually exist reduces slightly the direct stresses, and reduces still more slightly the deflections. See Fig. 1.

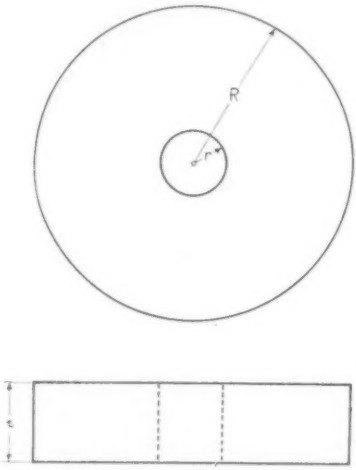


FIG. 1 TYPICAL HEAD WITH CENTRAL HOLE
(Any portion of head outside support circle not considered.)

d The loading is normal to the head, and may be so divided that each part is uniformly distributed over an annular zone. See Fig. 2. The three zones into which the head is divided by any such part of the loading will be called *inner zone*, *loaded zone* and *outer zone*. It is evident that this covers all cases of static fluid pressure in which the system is symmetrical with respect to all diameters. It covers also the case of a gasket load, for this may properly be regarded as uniformly distributed over a very narrow zone; in such a case, the width of the loaded zone is put equal to zero for the sake of convenience, although an absolute line contact is physically impossible. The same procedure as for a gasket load is applied, whenever there is a central hole, to a load concentrated around the edge of the hole, such as the force applied by an insert

or plug. In any case where a load is treated as if concentrated on a circle, the loaded zone is considered not to exist. For the typical load, which is uniformly distributed over an annular zone, the total shearing force on any normal cylindrical section concentric with the zones (hereafter called circumferential section), having radius x, may be represented by $S = m + nx^2$, where m and n are constants pertaining to the zone cut by the section.

15 For the sake of convenience, the head will be regarded as horizontal. Although the loading may act partly upward and

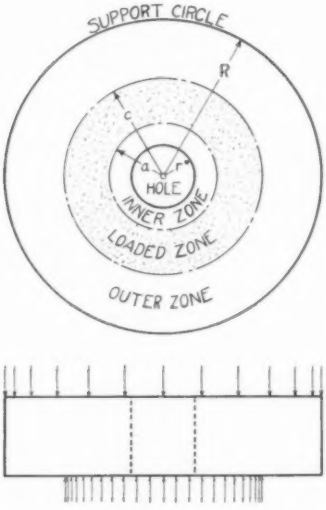


FIG. 2 DIAGRAM OF TYPICAL LOAD
(Uniformly distributed over annular zone of radii a and c.)

partly downward, the analysis contemplates only one part at a time, the joint effect of several loads being the aggregate of the several effects independently determined; it is therefore significant to distinguish between the opposite faces of the head as *tension face* and *compression face*, these terms having the same meaning as when applied to a simple beam.

16 Broadly considered, all loading is applied as normal external pressure, and causes an internal compressive force normal to the faces, throughout the loaded zone; the intensity of this force is equal to the intensity of applied pressure at the compression face, and decreases uniformly to zero at the tension face. If a load W be

distributed over an annular zone of area $\pi(c^2 - a^2)$, the intensity of normal compressive force at the compression face is

$$q'_n = -\frac{W}{\pi(c^2 - a^2)}$$

the negative sign indicating compression. There are exceptions to this rule of surface application (for example, the dead weight of the head), but they are unimportant because any errors due to disregarding them are very minute in any actual case; if it be desired to take into account these exceptions, those terms in the resulting formulae which depend on the value of q'_n may simply be dropped.

NOTATION

17 The following notation is used throughout the paper:

R = radius of support circle. Positive. Unit = Distance

c = outer radius of loaded zone. Positive. Unit = Distance

a = inner radius of loaded zone. Positive. Unit = Distance

r = radius of central hole, if any. Positive. Unit = Distance

t = thickness of head. Positive. Unit = Distance

x = variable distance from center. Positive. Unit = Distance

y = variable deflection of neutral surface due to flexure alone. Positive. Unit = Distance

Y = variable deflection of neutral surface due to detrusion alone. Positive. Unit = Distance

z = variable vertical ordinate of a point in the head, measured from neutral surface. Positive if measured in same direction as load, negative if opposite. Unit = Distance

W = total load on loaded zone. Positive. Unit = Force

E = modulus of elasticity (Young's). Positive. Unit = $\frac{\text{Force}}{\text{Dist. squared}}$

λ = Poisson's Ratio. Positive. Pure Number

S = variable total vertical shearing force on a circumferential section. Positive. Unit = Force

s = variable intensity of vertical or horizontal shearing force in a vertical radial plane, at any point in the head. Positive. Unit = $\frac{\text{Force}}{\text{Dist. squared}}$

s_m = Value of s at any point in neutral surface.

18 The symbols for the intensities of internal stress and of internal force, other than shear, are given in the following table:

	Tension Face			Compression Face		
	Radial	Tangent	Normal	Radial	Tangent	Normal
Direct Stress	p_r	p_t	p_n	p'_r	p'_t	p'_n
Direct Force	q_r	q_t	q_n	q'_r	q'_t	q'_n

The quantities represented in this table are positive if tensile, negative if compressive. Their common unit is Force ÷ Distance squared. Direct stress intensity is defined as the rate of direct strain multiplied by E.

19 For temporary convenience, the following two variables are introduced:

u = first derivative, with respect to x, of the bending moment acting on a vertical radial section; in other words, the intensity of tangential bending

v = first derivative, with respect to circumferential arc distance, of the bending moment acting on a circumferential section; in other words, the intensity of radial bending.

The variables u and v are positive if they tend toward concavity of the compression face, as defined; u is always positive, but v is negative near the support circle. Their common unit is Force × Distance ÷ Distance, or simply Force.

20 Although the deflections, which occur in the same direction as the load causing them, are considered positive, heed must be given to the fact that separate loads, acting in opposite directions, cause opposite deflections. Both y and Y are considered zero at the support circle.

21 The neutral surface is everywhere midway between the two faces, but its distance from either face is slightly affected by diminution of the thickness t due to surface application of loading. However, this effect is so exceedingly small, that it will be considered in this analysis to be quite negligible. Therefore the deflection of either face, for any value of x, will be regarded as identical with the deflection of the neutral surface for that value of x. The fact that detrusive strains cause curvature of the normal elements does not invalidate this proposition. No proofs are given for the statements contained in this paragraph, for they are based on general theory and are not peculiar to the problem in hand.

ANALYSIS OF FLEXURE AND DIRECT STRESS

22 From general theory,[1] the following six relations connecting stresses and forces are true for all values of x:

$$p_r = q_r - \lambda q_t - \lambda q_n \quad [1] \qquad p_r' = q_r' - \lambda q_t' - \lambda q_n' \quad [2]$$
$$p_t = q_t - \lambda q_r - \lambda q_n \quad [3] \qquad p_t' = q_t' - \lambda q_r' - \lambda q_n' \quad [4]$$
$$p_n = q_n - \lambda q_r - \lambda q_t \quad [5] \qquad p_n' = q_n' - \lambda q_r' - \lambda q_t' \quad [6]$$

23 The neutral surface is defined as the boundary between positive and negative intensities of direct force, or the locus of points at which no horizontal direct internal force exists. By the principle that straight elements normal to the neutral surface are not bent by flexure of the head, the neutral surface may be proved to lie midway between the faces; it follows that

$$q_r = -q_r' \quad [7] \qquad q_t = -q_t' \quad [8]$$

24 According to the last paragraph of the Preliminary Explanation, there is no direct internal force normal to the head at the tension face:

$$q_n = 0. \quad [9]$$

25 Substituting for q_r', q_t' and q_n in Eqs. [1] to [6], we have instead Eqs. [10] to [15], as follows:

$$p_r = q_r - \lambda q_t \quad [10] \qquad p_r' = -q_r + \lambda q_t - \lambda q_n' \quad [11]$$
$$p_t = q_t - \lambda q_r \quad [12] \qquad p_t' = -q_t + \lambda q_r - \lambda q_n' \quad [13]$$
$$p_n = -\lambda(q_r + q_t) \quad [14] \qquad p_n' = q_n' + \lambda(q_r + q_t) \quad [15]$$

26 From Eqs. [10] to [13], we derive expressions for q_t and q_r:

$$q_t = \frac{1}{2(1-\lambda^2)}[p_t - p_t' + \lambda(p_r - p'_r) - \lambda(1+\lambda)q_n'] \quad [16]$$

$$q_r = \frac{1}{2(1-\lambda^2)}[p_r - p_r' + \lambda(p_t - p_t') - \lambda(1+\lambda)q_n'] \quad [17]$$

$$q_t - q_r = \frac{1}{2(1+\lambda)}[p_t - p_t' - p_r + p_r'] \quad [18]$$

27 The couple formed by the tangential forces acting on a portion dx of any radial section is given by the expression[2]

$$\frac{t^2 q_t dx}{6}$$

because the force intensity varies uniformly from $+q_t$ at the tension face to $-q_t$ at the compression face. The expression given above is equal to $u dx$, according to the definition of u. Therefore

$$u = \frac{t^2 q_t}{6} \quad [19]$$

similarly,

$$v = \frac{t^2 q_r}{6} \quad [20]$$

[1] This theory, sometimes known as the Maximum Strain Theory, may be looked up in Lanza's Applied Mechanics.

[2] This expression is simply the section modulus of a rectangle, dx in width, and t in depth, multiplied by the force intensity at the most extreme distance from the neutral axis. See any standard textbook on mechanics of materials, such as Burr, Lanza, or Merriman.

28 In any radial section, the strain in a distance dx at the tension face is $\frac{p_r\,dx}{E}$ and at the compression face $\frac{p_r'\,dx}{E}$. Therefore, as illustrated by Fig. 3, the radius of curvature of a radial section of the neutral surface is $\frac{E \cdot t}{p_r - p_r'}$.[1] For any curve $y = f(x)$, the second derivative $\frac{d^2y}{dx^2}$ closely approximates the reciprocal of the radius of curvature wherever the first derivative $\frac{dy}{dx}$ is very small, the relative error of the approximation being $\frac{3}{2}\left(\frac{dy}{dx}\right)^2$. In the cases considered, this error is so exceedingly small as to be quite negligible. Paying due regard to sign conventions, we find

$$p_r - p_r' = -E \cdot t\frac{d^2y}{dx^2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [21]$$

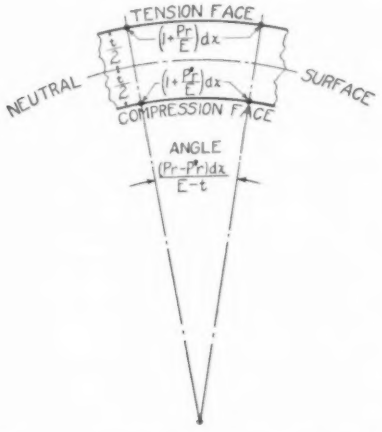


FIG. 3 CURVATURE IN RADIAL SECTION
(Note that p_r and p_r' must always have opposite signs.)

29 Any circumferential section, which before flexure is a right circular cylinder with axis passing through the center of the head, becomes due to flexure a right circular cone, the elements of which are normal to the neutral surface and hence are inclined to the vertical by the angle $\frac{dy}{dx}$. The trace of this cone

---

[1] Strictly, the radius of curvature $\frac{E \cdot t}{p_r - p'_r}$ pertains to the surface of zero radial strain rather than to the neutral surface. These two surfaces coincide throughout the unloaded zone (where $q_n' = 0$), but in the loaded zone they are separated by the vertical distance $\frac{t \cdot \lambda \cdot q_n'}{2(p_r - p_r')}$ due to surface application of load, the neutral surface having the greater radius of curvature). The relative error in the value for the radius of curvature, due to neglecting this distinction, is $\frac{t \cdot \lambda \cdot q_n'}{2(p_r - p_r')}$ divided by $\frac{E \cdot t}{p_r - p_r'}$, or $\frac{\lambda \cdot q_n'}{2E}$. This is evidently quite negligible.

on the tension face is a circle of radius $x\left(1+\frac{p_t}{E}\right)$, x having been the radius of the same circular element before strain, and p_t/E being by definition the rate of tangential strain. Likewise the trace of the cone on the compression face is a circle of radius $x\left(1+\frac{p_t'}{E}\right)$. The difference between the radii of these two circular traces, divided by the distance t, must equal the angle of inclination $\frac{dy}{dx}$ of the cone elements. See Fig. 4. Hence

$$p_t - p_t' = -\frac{Et}{x}\cdot\frac{dy}{dx} \quad \text{. . . . . [22]}$$

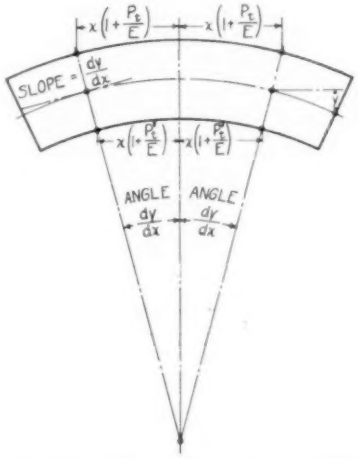


FIG. 4 CONE FORMED BY A CIRCUMFERENTIAL SECTION
(Note that p_t and p_t' must always have opposite signs.)

30 By means of Eqs. [21] and [22], we transform Eqs. [16] to [18] as follows:

$$q_t = \frac{E\cdot t}{2(1-\lambda^2)}\left[-\lambda\cdot\frac{d^2y}{dx^2}-\frac{1}{x}\cdot\frac{dy}{dx}\right]-\frac{\lambda\cdot q_n'}{2(1-\lambda)} \quad \text{. . . . . [23]}$$

$$q_r = \frac{E\cdot t}{2(1-\lambda^2)}\left[-\frac{d^2y}{dx^2}-\frac{\lambda}{x}\cdot\frac{dy}{dx}\right]-\frac{\lambda\cdot q_n'}{2(1-\lambda)} \quad \text{. . . . . [24]}$$

$$q_t - q_r = \frac{E\cdot t}{2(1+\lambda)}\left[\frac{d^2y}{dx^2}-\frac{1}{x}\cdot\frac{dy}{dx}\right] \quad \text{. . . . . [25]}$$

31 Differentiating Eq. [24] with respect to x, we have the following, since q_n' is not a function of x:

$$\frac{dq_r}{dx} = \frac{E\cdot t}{2(1-\lambda^2)}\left[-\frac{d^3y}{dx^3}-\frac{\lambda}{x}\cdot\frac{d^2y}{dx^2}+\frac{\lambda}{x^2}\cdot\frac{dy}{dx}\right] \quad \text{. . . . . [26]}$$

32 The equilibrium of any half annular segment of the head (see Fig. 5) requires that the moments due to u on the rectangular ends of the segment,

shall balance the moment due to shear and the moment due to v on the two curved faces of the segment. Let the vertical plane ON bisecting the segment be the plane parallel to which the components of the moments are taken. The moment due to u is $2udx$. Since $\frac{1}{2}S$ is the shear on a half circumferential section, and since the centroid of this shear is distant $\frac{2x}{\pi}$ from the center of the head, the couple due to shear is equal to the force $\frac{1}{2}S$ multiplied by the differential movement $\frac{2\,dx}{\pi}$ of its centroid, or $\frac{S\,dx}{\pi}$. If θ be a variable direction angle measured in either direction from ON, then the radial moment acting on a dif-

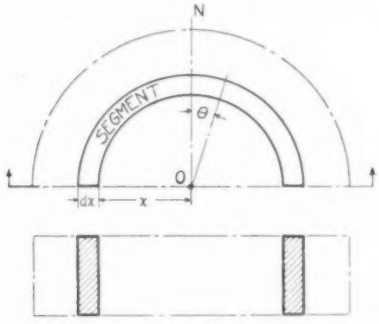


FIG. 5 EQUILIBRIUM OF HALF-ANNULAR SEGMENT

ferential portion $xd\theta$ of a circumferential section is $vxd\theta$, and has a component $vx \cdot \cos \theta d\theta$ parallel to the plane ON. The integral of this component between $\theta = -\frac{\pi}{2}$ and $\theta = +\frac{\pi}{2}$, when x is constant, is $2vx$. The expression $2vx$ is therefore the component parallel to ON of the total bending moment on the curved face of radius x. The corresponding moment for the opposite face of the segment differs from $2vx$ by the quantity $2d(vx)$, which is therefore the function of v affecting the equilibrium of the segment. Hence

$$u \cdot dx = d(vx) + \frac{S}{2\pi} \cdot dx \quad \ldots\ldots\ldots\ldots [27]$$

whence

$$u - v - x\frac{dv}{dx} = \frac{S}{2\pi} \quad \ldots\ldots\ldots\ldots [28]$$

By means of Eqs. [19] and [20], we may eliminate u and v from Eq. [28]:

$$q_t - q_r - x\frac{dq_r}{dx} = \frac{3S}{\pi t^2} \quad \ldots\ldots\ldots\ldots [29]$$

By means of Eqs. [25] and [26], we may eliminate q_t and q_r from Eq. [29]:

$$x \cdot \frac{d^3y}{dx^3} + \frac{d^2y}{dx^2} - \frac{1}{x} \cdot \frac{dy}{dx} = \frac{6(1 - \lambda^2)S}{\pi E t^3} \quad \ldots\ldots\ldots [30]$$

Dividing by x, and using the identity

$$\frac{1}{x} \cdot \frac{d^2y}{dx^2} - \frac{1}{x^2} \cdot \frac{dy}{dx} = \frac{d}{dx}\left(\frac{1}{x} \cdot \frac{dy}{dx}\right)$$

$$\frac{d^3y}{dx^3} + \frac{d}{dx}\left(\frac{1}{x} \cdot \frac{dy}{dx}\right) = \frac{6(1 - \lambda^2)S}{\pi \cdot E \cdot t^3 \cdot x} \quad \ldots\ldots\ldots [31]$$

Integrating and multiplying by x,

$$x \cdot \frac{d^2y}{dx^2} + \frac{dy}{dx} = \frac{6(1-\lambda^2)x}{\pi E t^3} \int_x \frac{S}{x} \quad \ldots\ldots\ldots [32]$$

Integrating again, since $x \cdot \frac{d^2y}{dx^2} + \frac{dy}{dx} = \frac{d}{dx}\left(x \cdot \frac{dy}{dx}\right)$ and dividing by x,

$$\frac{dy}{dx} = \frac{6(1-\lambda^2)}{\pi E t^3} \cdot \frac{1}{x} \int_x x \int_x \frac{S}{x} \quad \ldots\ldots\ldots [33]$$

Integrating again,

$$y = \frac{6(1-\lambda^2)}{\pi E t^3} \int_x \frac{1}{x} \int_x x \int_x \frac{S}{x} \quad \ldots\ldots\ldots [34]$$

33 From d Par. 14 of the initial assumptions, $S = m + nx^2$, where m and n are constants determined by the loading and having different values for different zones. Substituting for S in Eq. [**34**], performing the triple integration of the right-hand member, and transforming the three undetermined constants of integration in such manner as to suit convenience later, the following general equation for the flexure deflection is derived: [1]

$$y = \frac{3(1-\lambda^2)W}{4\pi E t^3}\left[A \log_e x + B + \frac{2m}{W} x^2 \cdot \log_e x - Cx^2 + \frac{n}{4W} x^4\right] - \frac{\lambda q_n'}{2Et} x^2 \quad [35]$$

34 The constant factor W and the term containing q_n' do not appear naturally in Eq. [**35**], but their introduction has a favorable effect on the expressions later derived for the arbitrary constants A, B, C; since q_n', like m and n, is a constant throughout any zone, the term containing q_n' merely affects the value of the constant C.

35 From Eq. [**35**], by differentiation, we obtain:

$$\frac{dy}{dx} = \frac{3(1-\lambda^2)W}{4\pi E t^3}\left[\frac{A}{x} + \frac{4m}{W} x \cdot \log_e x + \frac{2m}{W} x - 2Cx + \frac{n}{W} x^3\right] - \frac{\lambda \cdot q_n'}{Et} x. \quad [36]$$

$$\frac{d^2y}{dx^2} = \frac{3(1-\lambda^2)W}{4\pi E \cdot t^3}\left[-\frac{A}{x^2} + \frac{4m}{W} \log_e x + \frac{6m}{W} - 2C + \frac{3n}{W} x^2\right] - \frac{\lambda \cdot q_n'}{E \cdot t} \quad [37]$$

---

[1] The steps in the triple integration and simplification between Equations 34 and 35 are as follows:

$$y = \frac{6(1-\lambda^2)}{\pi \cdot E \cdot t^3} \int_x \frac{1}{x} \int_x x \int_x \left(\frac{m}{x} + nx\right)$$

$$= \frac{6(1-\lambda^2)}{\pi E t^3} \int_x \frac{1}{x} \int_x x \left(m \cdot \log_e x + \frac{nx^2}{2} + C'\right)$$

$$= \frac{6(1-\lambda^2)}{\pi E t^3} \int_x \frac{1}{x} \left(\frac{mx^2}{2} \log_e x - \frac{mx^2}{4} + \frac{nx^4}{8} + \frac{C'x^2}{2} + A'\right)$$

$$= \frac{6(1-\lambda^2)}{\pi E t^3} \left(\frac{mx^2 \log_e x}{4} - \frac{mx^2}{4} + \frac{nx^4}{32} + \frac{C'x^2}{4} + A' \log_e x + B'\right)$$

$$= \frac{3(1-\lambda^2)}{4\pi E t^3}\left[8A' \log_e x + 8B' + 2mx^2 \log_e x - 2x^2(m - C') + \frac{nx^4}{4}\right]$$

$$= \frac{3(1-\lambda^2)}{4\pi E t^3}\left[8A' \log_e x + 8B' + 2mx^2 \log_e x - 2x^2\left(m - C' - \frac{\pi t^2 \lambda q_n'}{3(1-\lambda^2)}\right) + \frac{nx^4}{4}\right] - \frac{\lambda q_n'}{2Et} x^2$$

$$= \frac{3(1-\lambda^2)W}{4\pi E t^3}\left[\frac{8A'}{W} \log_e x + \frac{8B'}{W} + \frac{2m}{W} x^2 \log_e x - \frac{2x^2}{W}\left(m - C' - \frac{\pi t^2 \lambda q_n'}{3(1-\lambda^2)}\right) + \frac{n}{4W} x^4\right] - \frac{\lambda q_n'}{2Et} x^2$$

$$= \frac{3(1-\lambda^2)W}{4\pi E t^3}\left[A \log_e x + B + \frac{2m}{W} x^2 \log_e x - Cx^2 + \frac{n}{4W} x^4\right] - \frac{\lambda q_n'}{2Et} \cdot x^2$$

36 Substituting these values for the first and second derivatives in Eqs. [23] and [24], we obtain the following expressions for q_t and q_r:

$$q_t = -q_t' = \frac{3W}{8\pi t^2}\left[-\frac{A}{x^2}(1-\lambda)-\frac{4m}{W}\log_e x(1+\lambda)-\frac{2m}{W}(1+3\lambda)+2C(1+\lambda)-\frac{nx^2}{W}(1+3\lambda)\right] \quad [38]$$

$$q_r = -q_r' = \frac{3W}{8\pi t^2}\left[+\frac{A}{x^2}(1-\lambda)-\frac{4m}{W}\log_e x(1+\lambda)-\frac{2m}{W}(3+\lambda)+2C(1+\lambda)-\frac{nx^2}{W}(3+\lambda)\right] \quad [39]$$

37 Substituting these values for q_t and q_r in Eqs. [10] to [13], we obtain the following expressions for p_t and p_r:

$$p_t = -p_t' - \lambda q_n' = \frac{3(1-\lambda^2)W}{8\pi \cdot t^3}\left[-\frac{A}{x^2}-\frac{4m}{W}\log_e x-\frac{2m}{W}+2C-\frac{nx^2}{W}\right] \quad [40]$$

$$p_r = -p_r' - \lambda q_n' = \frac{3(1-\lambda^2)W}{8\pi \cdot t^3}\left[+\frac{A}{x^2}-\frac{4m}{W}\log_e x-\frac{6m}{W}+2C-\frac{3nx^2}{W}\right] \quad [41]$$

38 Each of the three zones has its own constants m, n, q_n'. The values of m and n for any zone are found by inspection of the shear equation for that zone, the shear S for any value of x being merely the amount of load included within the circle of radius x. The surface pressure q_n' is evidently zero for inner and outer zones, and has the value $-\frac{W}{\pi(c^2-a^2)}$ for the loaded zone. Thus:

Inner zone: $S = m + nx^2 = 0.$ $\quad m = 0,$ $\quad n = 0,$ $\quad q_n' = 0.$

Loaded zone: $S = m + nx^2 = W\frac{x^2-a^2}{c^2-a^2}; m = -\frac{a^2W}{c^2-a^2}; n = +\frac{W}{c^2-a^2}; q_n' = -\frac{W}{\pi(c^2-a^2)}$

Outer zone: $S = m + nx^2 = W$ $\quad m = W$ $\quad n = 0$ $\quad q_n' = 0$

39 Substituting these values in Eqs. [35], [38], [39], [40] and [41], the following equations are deduced for the three different zones:

Inner zone:

$$y = \frac{3(1-\lambda^2)W}{4\pi E \cdot t^3}\left[A_1\log_e x + B_1 - C_1x^2\right] \quad [42]$$

$$q_t = -q_t' = \frac{3W}{8\pi t^2}\left[-\frac{A_1}{x^2}(1-\lambda)+2C_1(1+\lambda)\right] \quad [43]$$

$$q_r = -q_r' = \frac{3W}{8\pi t^2}\left[+\frac{A_1}{x^2}(1-\lambda)+2C_1(1+\lambda)\right] \quad [44]$$

$$p_t = -p_t' = \frac{3(1-\lambda^2)W}{8\pi t^3}\left[-\frac{A_1}{x^2}+2C_1\right] \quad [45]$$

$$p_r = -p_r' = \frac{3(1-\lambda^2)W}{8\pi t^3}\left[+\frac{A_1}{x^2}+2C_1\right] \quad [46]$$

Loaded zone:

$$y = \frac{3(1-\lambda^2)W}{4\pi E \cdot t^3}\left[A_2\log_e x + B_2 - \frac{2a^2}{c^2-a^2}x^2\log_e x - C_2x^2 + \frac{x^4}{4(c^2-a^2)}\right] + \frac{\lambda \cdot W \cdot x^2}{2\pi Et(c^2-a^2)} \quad (47)$$

$$q_t = -q_t' = \frac{3W}{8\pi t^2}\left[-\frac{A^2}{x^2}(1-\lambda)+\frac{4a^2}{c^2-a^2}\log_e x(1+\lambda)+\frac{2a^2}{c^2-a^2}(1+3\lambda)+2C_2(1+\lambda)-\frac{x^2}{c^2-a^2}(1+3\lambda)\right] \quad [48]$$

$$q_r = -q_r' = \frac{3W}{8\pi t^2}\left[+ \frac{A_2}{x^2}(1-\lambda) + \frac{4a^2}{c^2-a^2}\log_e x(1+\lambda) + \frac{2a^2}{c^2-a^2}(3+\lambda) + 2C_2(1+\lambda) - \frac{x^2}{c^2-a^2}(3+\lambda)\right] \quad [49]$$

$$p_t = -p_t' + \frac{\lambda W}{\pi(c^2-a^2)} = \frac{3(1-\lambda^2)W}{8\pi \cdot t^2}\left[- \frac{A_2}{x^2} + \frac{4a^2}{c^2-a^2}\log_e x + \frac{2a^2}{c^2-a^2} + 2C_2 - \frac{x^2}{c^2-a^2}\right] \quad [50]$$

$$p_r = -p_r' + \frac{\lambda W}{\pi(c^2-a^2)} = \frac{3(1-\lambda^2)W}{8\pi \cdot t^2}\left[+ \frac{A_2}{x^2} + \frac{4a^2}{c^2-a^2}\log_e x + \frac{6a^2}{c^2-a^2} + 2C_2 - \frac{3x^2}{c^2-a^2}\right] \quad [51]$$

Outer zone:

$$y = \frac{3(1-\lambda^2)W}{4\pi E \cdot t^3}\left[A_3 \log_e x + B_3 + 2x^2 \cdot \log_e x - C_3 x^2\right] \quad [52]$$

$$q_t = -q_t' = \frac{3W}{8\pi t^2}\left[- \frac{A_3}{x^2}(1-\lambda) - 4\log_e x(1+\lambda) - 2(1+3\lambda) + 2C_3(1+\lambda)\right] \quad [53]$$

$$q_r = -q_r' = \frac{3W}{8\pi t^2}\left[+ \frac{A_3}{x^2}(1-\lambda) - 4\log_e x(1+\lambda) - 2(3+\lambda) + 2C_3(1+\lambda)\right] \quad [54]$$

$$p_t = -p_t' = \frac{3(1-\lambda^2)W}{8\pi t^2}\left[- \frac{A_3}{x^2} - 4\log_e x - 2 + 2C_3\right] \quad [55]$$

$$p_r = -p_r' = \frac{3(1-\lambda^2)W}{8\pi t^2}\left[+ \frac{A_3}{x^2} - 4\log_e x - 6 + 2C_3\right] \quad [56]$$

DETERMINATION OF CONSTANTS OF INTEGRATION

40 There are nine constants of integration, three for each zone, as yet undetermined. Physical conditions fix these values, nine equations of condition being deduced as follows.

41 By definition, $y = 0$ at the support circle. Hence by Eq. [52],

$$A_3 \cdot \log_e R + B_3 + 2R^2 \cdot \log_e R - C_3 \cdot R^2 = 0 \quad [I]$$

42 Since there can be no step or offset in either face, the neutral surface, lying midway between faces, cannot be stepped; therefore y is continuous throughout. Hence Eqs. [52] and [47] must yield identical values for y at $x = c$, and Eqs. [42] and [47] must yield identical values at $x = a$. Substituting and rearranging,

$$(A_3-A_2)\cdot\log_e c + (B_3-B_2) - (C_3-C_2)\cdot c^2 = -\frac{c^4}{c^2-a^2}\left(2\log_e c - \frac{1}{4}\right) + \frac{2\lambda \cdot t^2 c^2}{3(1-\lambda^2)(c^2-a^2)}. \quad [II]$$

$$(A_1-A_2)\cdot\log_e a + (B_1-B_2) - (C_1-C_2)\cdot a^2 = -\frac{a^4}{c^2-a^2}\left(2\log_e a - \frac{1}{4}\right) + \frac{2\lambda\cdot t^2 a^2}{3(1-\lambda^2)(c^2-a^2)}. \quad [III]$$

43 Equilibrium requires that the force intensities q_t and q_r be continuous throughout; i.e., these quantities cannot change abruptly at zone boundaries. Hence Eqs. [53] and [48] must coincide at $x = c$, Eqs. [43] and [48] at $x = a$, Eqs. [54] and [49] at $x = c$, and Eqs. [44] and [49] at $x = a$.

$$-(A_3 - A_2)\frac{1-\lambda}{c^2} + 2(C_3 - C_2)(1+\lambda) = \frac{c^2}{c^2-a^2}\left[4\log_e c(1+\lambda) + 1 + 3\lambda\right]. \quad [IV]$$

$$-(A_1 - A_2)\frac{1-\lambda}{a^2} + 2(C_1 - C_2)(1+\lambda) = \frac{a^2}{c^2-a^2}\left[4\log_e a(1+\lambda) + 1 + 3\lambda\right]. \quad [V]$$

$$+ (A_3 - A_2)\frac{1-\lambda}{c^2} + 2(C_3 - C_2)(1+\lambda) = \frac{c^2}{c^2 - a^2}\left[4 \log_e c(1+\lambda) + 3 + \lambda\right] \quad . \quad [\text{VI}]$$

$$+ (A_1 - A_2)\frac{1-\lambda}{a^2} + 2(C_1 - C_2)(1+\lambda) = \frac{a^2}{c^2 - a^2}\left[4 \log_e a(1+\lambda) + 3 + \lambda\right] . \quad [\text{VII}]$$

44 The radial force intensity q_r must be zero at $x = R$, for the head is simply supported at that circle. Hence, by Eq. [**54**],

$$\frac{A_3}{R^2}(1-\lambda) - 4 \log_e R(1+\lambda) - 2(3+\lambda) + 2C_3(1+\lambda) = 0. \quad . \quad [\text{VIII}]$$

45 The ninth condition depends on whether or not there is a central hole through the head. A solid head is not a special case of a head with a hole, for the stresses and forces at the edge of such a hole do not approach as limits the corresponding quantities for a solid head when the radius of the hole approaches zero. The reason for this is that in case of a hole the radial restraint at the center is lacking. This explains why a solid head cannot be treated as a head having a central hole of zero radius, and why a hole, even if very small, causes maximum stresses far in excess of the corresponding stresses in a solid head. The remainder of the analysis for flexure and direct stresses considers two classes of heads, viz., solid heads and heads centrally bored. In case of a solid head the slope $\frac{dy}{dx}$ must be zero at $x = 0$, whereas, in case of a hole, the radial direct force must be of zero intensity at the edge of the hole. Thus the ninth equation of condition must be one of two alternative equations.

a For solid head, $\frac{dy}{dx} = 0$ at $x = 0$. Hence, by Eq. [**36**] $\frac{A_1}{x} - 2C_1 \cdot x = 0$ for $x = 0$; since C_1 must be finite, $\frac{A_1}{x} = 0$ for $x = 0$, and $A_1 = 0$. *b* For head with central hole of radius r, $q_r = 0$ at $x = r$, whence by Eq. [**44**], $\frac{A_1}{r^2}(1-\lambda) + 2C_1(1+\lambda) = 0$. Therefore in any case the ninth equation is one of the following:

For solid head, $$A_1 = 0. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [\text{IXa}]$$

For head with hole, $$A_1 + 2r^2 \cdot C_1 \cdot \frac{1+\lambda}{1-\lambda} = 0. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [\text{IXb}]$$

46 Solving the equations of condition, we obtain the following:

Solid head:

Inner zone

$$\begin{cases} A_1 = 0 \quad . \quad [\textbf{57}] \\ B_1 = \frac{3+\lambda}{1+\lambda}R^2 - (c^2+a^2)\left(\log_e R + \frac{3}{4} + \frac{1}{1+\lambda}\right) + \frac{1}{c^2-a^2}(c^4 \log_e c - a^4 \log_e a) - \frac{2 \cdot t^2}{3(1-\lambda^2)} \quad [\textbf{58}] \\ C_1 = 2 \log_e R - \frac{1-\lambda}{1+\lambda} \cdot \frac{c^2+a^2}{2R^2} - \frac{2}{c^2-a^2}(c^2 \cdot \log_e c - a^2 \log_e a) + \frac{2}{1+\lambda} \quad [\textbf{59}] \end{cases}$$

Loaded zone

$$\begin{cases} A_2 = -\frac{a^4}{c^2-a^2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [\textbf{60}] \\ B_2 = \frac{3+\lambda}{1+\lambda}R^2 - (c^2+a^2)\left(\log_e R - \frac{1}{2} + \frac{1}{1+\lambda}\right) + \frac{c^4}{c^2-a^2}\left(\log_e c - \frac{5}{4}\right) - \frac{2\lambda \cdot t^2 c^2}{3(1-\lambda^2)(c^2-a^2)} \quad [\textbf{61}] \\ C_2 = 2 \log_e R - \frac{1-\lambda}{1+\lambda} \cdot \frac{c^2+a^2}{2R^2} - \frac{c^2}{c^2-a^2}(2 \log_e c + 1) + \frac{3+\lambda}{1+\lambda} \quad . \quad . \quad . \quad [\textbf{62}] \end{cases}$$

Outer zone

$$A_3 = c^2 + a^2 \quad [63]$$

$$B_3 = \frac{3+\lambda}{1+\lambda} R^2 - (c^2 + a^2)(\log_e R - \frac{1}{2} + \frac{1}{1+\lambda} \quad [64]$$

$$C_3 = 2 \log_e R - \frac{1-\lambda}{1+\lambda} \cdot \frac{c^2+a^2}{2R^2} + \frac{3+\lambda}{1+\lambda} \quad [65]$$

Head with central hole of radius r:

Inner zone

$$A_1 = -\frac{2r^2}{1-\frac{r^2}{R^2}} \cdot \frac{1+\lambda}{1-\lambda}\left[2\log_e R - \frac{1-\lambda}{1+\lambda}\cdot\frac{c^2+a^2}{2R^2} - \frac{2}{c^2-a^2}\left(c^2\log_e c - a^2\log_e a\right) + \frac{2}{1+\lambda}\right] \quad [66]$$

$$B_1 = (C_1+1)R^2 - (A_1+2R^2)\log_e R - (c^2+a^2)\left(\log_e R + \frac{5}{4}\right) + \frac{1}{c^2-a^2}\left[c^4\log_e c - a^4\log_e a + 2R^2(c^2\log_e c - a^2\log_e a)\right] - \frac{2\lambda\cdot t^2}{3(1-\lambda^2)} \quad [67]$$

$$C_1 = \frac{1}{1-\frac{r^2}{R^2}}\left[2\log_e R - \frac{1-\lambda}{1+\lambda}\cdot\frac{c^2+a^2}{2R^2} - \frac{2}{c^2-a^2}(c^2\cdot\log_e c - a^2\cdot\log_e a) + \frac{2}{1+\lambda}\right] \quad [68]$$

Loaded zone

$$A_2 = -\frac{a^4}{c^2-a^2} - \frac{2r^2}{1-\frac{r^2}{R^2}} \cdot \frac{1+\lambda}{1-\lambda}\left[2\log_e R - \frac{1-\lambda}{1+\lambda}\cdot\frac{c^2+a^2}{2R^2} - \frac{2}{c^2-a^2}(c^2\log_e c - a^2\log_e a) + \frac{2}{1+\lambda}\right] \quad [69]$$

$$B_2 = C_2\cdot R^2 - (A_2+2R^2)\log_e R + \frac{c^2}{c^2-a^2}\left[R^2 - c^2\left(\log_e R + \frac{5}{4}\right) + \log_e c(2R^2+c^2)\right] - \frac{2\lambda\cdot t^2c^2}{3(1-\lambda^2)(c^2-a^2)} \quad [70]$$

$$C_2 = \frac{1}{1-\frac{r^2}{R^2}}\left[2\log_e R - \frac{1-\lambda}{1+\lambda}\cdot\frac{c^2+a^2}{2R^2} - \frac{c^2}{c^2-a^2}(2\log_e c + 1) + \frac{a^2r^2}{(c^2-a^2)R^2}(2\log_e a + 1) + \frac{3+\lambda}{1+\lambda}\right] \quad [71]$$

Outer zone

$$A_3 = c^2+a^2 - \frac{2r^2}{1-\frac{r^2}{R^2}}\cdot\frac{1+\lambda}{1-\lambda}\left[2\log_e R - \frac{1-\lambda}{1+\lambda}\cdot\frac{c^2+a^2}{2R^2} - \frac{2}{c^2-a^2}(c^2\cdot\log_e c - a^2\cdot\log_e a) + \frac{2}{1+\lambda}\right] \quad [72]$$

$$B_3 = C_3\cdot R^2 - (A_3+2R^2)\log_e R \quad [73]$$

$$C_3 = \frac{1}{1-\frac{r^2}{R^2}}\left[2\log_e R - \frac{1-\lambda}{1+\lambda}\cdot\frac{c^2+a^2}{2R^2} - \frac{2r^2}{(c^2-a^2)R^2}(c^2\cdot\log_e c - a^2\cdot\log_e a) + \frac{2}{1+\lambda}\right] + 1 \quad [74]$$

47 In case the loaded zone is so narrow that the application of load is through "line contact," we may consider that the loaded zone disappears as such, becoming a loaded boundary line between inner and outer zones; we may then transform the equations pertaining to inner and outer zones, by finding the limits which they approach when a approaches c as a limit. This procedure is in error, in that the assumption of infinitesimal width of zone contradicts the basic presumption that the elastic limit for the material is not exceeded in any way; but the approximation is very close indeed, and is a practical neces-

sity, because reliable information as to the actual width of a line contact cannot generally be obtained. Owing to the inconsistency, the quantities y, q_t, q_t', q_r, q_r' lose their continuity across the zone boundary, slight breaks appearing in their values due to the assumption of a material force concentrated on an infinitesimal area. The value of y is found to increase abruptly by the amount $\frac{\lambda W}{2\pi E t}$ as x increases past the boundary; the value of y at the boundary is therefore not a single quantity according to the equations, but, as the break is in any case very small, no considerable error can be involved if the midway value is used. The only loads of this kind commonly occurring, are gasket and joint reactions and central plug reactions.

48 As Eqs. [57] to [74] do not directly yield values in the case of a concentrated load, it is necessary to transform those pertaining to inner and outer zones by finding the limits which the expressions approach when a approaches c as its limit. The equations for the loaded zone are disregarded. The results are as follows:

Solid head, load concentrated on circle of radius c:

Inner zone

$$A_1 = 0 \quad [75]$$

$$B_1 = \frac{3+\lambda}{1+\lambda}\left(R^2 - c^2\right) - 2c^2 \cdot \log_e \frac{R}{c} - \frac{2\lambda t^2}{3(1-\lambda^2)} \quad [76]$$

$$C_1 = 2\log_e \frac{R}{c} + \frac{1-\lambda}{1+\lambda}\left(1 - \frac{c^2}{R^2}\right) \quad [77]$$

Outer zone

$$A_3 = 2c^2 \quad [78]$$

$$B_3 = \frac{3+\lambda}{1+\lambda}R^2 - 2c^2\left(\log_e R - \frac{1}{2} + \frac{1}{1+\lambda}\right) \quad [79]$$

$$C_3 = 2\log_e R + 2 + \frac{1-\lambda}{1+\lambda}\left(1 - \frac{c^2}{R^2}\right) \quad [80]$$

Head with hole, load concentrated on circle of radius c:

Inner zone

$$A_1 = -\frac{2r^2}{1-\frac{r^2}{R^2}}\left[2\frac{1+\lambda}{1-\lambda}\log_e \frac{R}{c} + 1 - \frac{c^2}{R^2}\right] \quad [81]$$

$$B_1 = C_1 \cdot R^2 - A_1 \cdot \log_e R - 2(R^2 + c^2)\log_e \frac{R}{c} + 2(R^2 - c^2) - \frac{2\lambda \cdot t^2}{3(1-\lambda^2)} \quad [82]$$

$$C_1 = \frac{1}{1-\frac{r^2}{R^2}}\left[2\log_e \frac{R}{c} + \frac{1-\lambda}{1+\lambda}\left(1 - \frac{c^2}{R^2}\right)\right] \quad [83]$$

Outer zone

$$A_3 = 2c^2 - \frac{2r^2}{1-\frac{r^2}{R^2}}\left[2\frac{1+\lambda}{1-\lambda}\log_e \frac{R}{c} + 1 - \frac{c^2}{R^2}\right] \quad [84]$$

$$B_3 = C_3 \cdot R^2 - (A_3 + 2R^2)\log_e \cdot R \quad [85]$$

$$C_3 = \frac{1}{1-\frac{r^2}{R^2}}\left[2\log_e R + 2 + \frac{1-\lambda}{1+\lambda}\left(1 - \frac{c^2}{R^2}\right) - \frac{2r^2}{R^2}(\log_e c + 1)\right] \quad [86]$$

49 This completes the general analysis for flexure and direct stress. Eqs. [42] to [56], together with the formulas for the constants of integration, provide means for determining flexure strains, direct internal forces and direct stresses at all points. Some important special formulas will now be deduced from the general equations.

SPECIAL FORMULAS

50 In a solid head, the greatest tensile stress due to any load W is $p_t = p_r$ at the center; except for an extraordinary combination of loads in opposite directions, the greatest resultant tensile stress in the head is the resultant at the center. The greatest compressive stress due to any load W is $p_t' = p_r'$ at the center, unless W is applied over a very small area including the center, in which case the greatest compressive stress may be p_t' (not p_r') at the edge of the loaded area; except for an extraordinary combination of loads in opposite directions, the greatest resultant compressive stress is the resultant at the center (or very near the center, as contemplated above). The greatest deflection due to flexure is y at the center, except for very extraordinary combinations of loads in opposite directions; the value for y given by the formulas, applies to both faces as well as to the neutral surface.

51 In a head with a central hole, due to any load W, the greatest tensile stress is p_t at the edge of the hole, the greatest compressive stress is p_t' at the edge of the hole (except for a load so concentrated on a narrow zone around the edge of the hole that the greatest stress is p_t' at the outer boundary of the zone), and the greatest deflection is y at the edge of the hole. Extraordinary combinations of loads in opposite directions may cause the greatest resultant stresses to occur elsewhere than at the edge of the hole, and may even shift the maximum resultant deflection away from the hole. At and near the edge of the hole, due to any single load W, p_r has a negative value and is therefore compressive, and p_r' has a positive value and is therefore tensile.

52 For a solid head, conditions at the center are as follows:

$$y = \frac{3(1-\lambda^2)W \cdot B_1}{4\pi E \cdot t^3} \quad [87]$$

$$q_t = q_r = -q_t' = -q_r' = \frac{3(1+\lambda)W \cdot C_1}{4\pi \cdot t^2} \quad [88]$$

$$p_t = p_r = \frac{3(1-\lambda^2)W \cdot C_1}{4\pi \cdot t^2} \quad [89]$$

$$p_t' = p_r' = -\frac{3(1-\lambda^2)W \cdot C_1}{4\pi \cdot t^2} - \lambda \cdot q_n' \quad [90]$$

where $q_n' = 0$ if the load does not cover the center,

but $q_n' = -\frac{W}{\pi \cdot C^2}$ if the loaded zone extends to the center, i.e., if $a = 0$.

53 For a head with a central hole of radius r, conditions at the edge of the hole are as follows:

$$y = \frac{3(1-\lambda^2)W}{4\pi \cdot E \cdot t^3}\left[A_1 \cdot \log_e r + B_1 - C_1 r^2\right] \quad [91]$$

$$q_t = -q_t' = \frac{3W}{8\pi \cdot t^2}\left[-\frac{A_1}{r^2}(1-\lambda) + 2C_1(1+\lambda)\right] \quad [92]$$

$$q_r = -q_r' = \frac{3W}{8\pi \cdot t^2}\left[+\frac{A_1}{r^2}(1-\lambda) + 2C_1(1+\lambda)\right] \quad [93]$$

$$p_t = \frac{3(1-\lambda^2)W}{8\pi \cdot t^2}\left[-\frac{A_1}{r^2} + 2C_1\right] \quad [94]$$

$$p_t' = -\frac{3(1-\lambda^2)W}{8\pi \cdot t^2}\left[-\frac{A_1}{r^2} + 2C_1\right] - \lambda \cdot q_n' \quad [95]$$

$$p_r = \frac{3(1-\lambda^2)W}{8\pi \cdot t^2}\left[+\frac{A_1}{r^2} + 2C_1\right] \quad [96]$$

$$p_r' = -\frac{3(1-\lambda^2)W}{8\pi \cdot t^2}\left[+\frac{A_1}{r^2} + 2C_1\right] - \lambda \cdot q_n' \quad [97]$$

In Eqs. [95] and [97] $q_n' = 0$ if the load does not extend to the edge of the hole, but $q_n' = -\frac{W}{\pi(c^2 - a^2)}$ if the load extends to the edge of the hole, i.e., if $a = r$.

54 The case of a solid disk uniformly loaded over the entire area of the support circle is not usual in the class of work considered, but yields very simple equations which may be useful in some preliminary investigations. In this case the inner and outer zones do not appear, for $a = 0$ and $c = R$.

$$y = \frac{W}{\pi \cdot R^2} \cdot \frac{R^2 - x^2}{2Et}\left[\frac{3(1-\lambda^2)}{8t^2}\left(\frac{5+\lambda}{1+\lambda}R^2 - x^2\right) - \lambda\right] \quad [98]$$

$$q_t = -q_t' = \frac{W}{\pi \cdot R^2} \cdot \frac{3(3+\lambda)}{8t^2}\left[R^2 - \frac{1+3\lambda}{3+\lambda}x^2\right] \quad [99]$$

$$q_r = -q_r' = \frac{W}{\pi \cdot R^2} \cdot \frac{3(3+\lambda)}{8t^2}[R^2 - x^2] \quad [100]$$

$$p_t = -p_t' + \frac{\lambda W}{\pi R^2} = \frac{W}{\pi \cdot R^2} \cdot \frac{3(1-\lambda^2)}{8t^2}\left[\frac{3+\lambda}{1+\lambda}R^2 - x^2\right] \quad [101]$$

$$p_r = -p_r' + \frac{\lambda W}{\pi R^2} = \frac{W}{\pi \cdot R^2} \cdot \frac{3(1-\lambda^2)}{8t^2}\left[\frac{3+\lambda}{1+\lambda}R^2 - 3x^2\right] \quad [102]$$

55 At the center of such a disk, conditions are as follows:

$$y = \frac{W}{2\pi Et}\left[\frac{3(1-\lambda)(5+\lambda)R^2}{8t^2} - \lambda\right] \quad [103]$$

$$q_t = q_r = -q_t' = -q_r' = \frac{3(3+\lambda)W}{8\pi t^2} \quad [104]$$

$$p_t = p_r = -p_t' + \frac{\lambda W}{\pi R^2} = -p_r' + \frac{\lambda W}{\pi R^2} = \frac{3(1-\lambda)(3+\lambda)W}{8\pi \cdot t^2} \quad [105]$$

56 Inasmuch as the stresses at the edge of the hole, in the case of a head having a *small* hole, do not differ greatly from the limits which their values would approach if r should approach zero, it is useful to analyze the conditions at the edge of the hole, in the hypothetical case of a head with an infinitesimal hole. The deflection is the same as for a solid head. The expressions for forces and stresses at the edge of the hole are found by substituting for A_1 and C_1 (as given in Eqs. [66] and [68], in Eqs. [92] to [97], and then putting $r = 0$. This procedure leads to the following expressions, in which C_1 is as given by Eq. [59] (or by Eq. [68] with r put equal to zero).

$$q_t = -q_t' = \frac{3(1+\lambda)W \cdot C_1}{2\pi \cdot t^2} \quad [106]$$

$$q_r = -q_r' = 0 \quad [107]$$

$$p_t = -p_t' - \lambda \cdot q_n' = \frac{3(1+\lambda)W \cdot C_1}{2\pi \cdot t^2} \quad [108]$$

$$p_r = -p_r' - \lambda \cdot q_n' = -\frac{3\lambda(1+\lambda)W \cdot C_1}{2\pi \cdot t^2} \quad [109]$$

57 It may be observed that q_t by Eq. [106] is double the value for q_t at the center of a solid head, and that p_t by Eq. [108] is $\frac{2}{1-\lambda}$ times the value of p_t at the center of a solid head. In the case of a very small hole, the high intensity of tangential stress is local, as p_t diminishes very rapidly when x increases away from the hole; at a short distance from such a hole, the stresses are not very different from the stresses at the same value of x in the case of a solid head. In the hypothetical limiting case contemplated in Eqs. [105] to [109], where r is infinitesimal, it may be shown that in an infinitesimal distance dx, measured radially from the edge of the infinitesimal hole, p_t drops abruptly to the value it would have at the center if the head were solid. Hence there occurs a right-angle break in the graph of p_t vs. x at $x = 0$, if a hole of zero radius be assumed to exist. This demonstrates the connecting link between a solid head and one with an indefinitely small hole.

58 A special case of occasional interest is that of a load concentrated on a very small circular space at the center of the head, the head being solid. The flexure deflection due to such a load approaches a finite limit for any given value of x, when the radius c of the loaded space approaches zero, and the forces and stresses approach finite limits for any value of x other than zero; but the force and stress values at the center increase without limit when the loaded area contracts. Actually, of course, the load requires a material space of application. Therefore, not only from the standpoint of the mathematical analysis, but also from physical considerations, it is necessary to assign a material value to the dimension c in order to calculate the force and stress values at the center. In other words, it is impossible to recognize a *point application* of load for purposes of stress analysis.

ANALYSIS OF SHEARING STRESS AND DETRUSIVE DEFLECTION

59 In any circumferential section of radius x, the external shearing force per unit of circumference is $\frac{S}{2\pi x}$. This must be numerically equal to the integral of the variable intensity of internal shear s, over unit horizontal distance and over a vertical range from $z = -\frac{1}{2}t$ to $z = +\frac{1}{2}t$, or

$$\frac{S}{2\pi x} = \int_{-\frac{t}{2}}^{+\frac{t}{2}} s \cdot dz \quad \ldots\ldots\ldots\ldots \quad [110]$$

60 Internal equilibrium, governing the case of an elementary cube of the material, requires that for any given value of x, the derivative $\frac{ds}{dz}$ shall be directly proportional to the horizontal intensity of radial direct force; this intensity of radial force, according to the general theory of flexure, is directly proportional to z. Hence, for any given value of x,

$$\frac{ds}{dz} = kz \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad [111]$$

where k is a constant.

61 By Eq. [111] since s_m is the value of s for $z = 0$,

$$s = s_m + \int_0^z k \cdot z \cdot dz = s_m + \frac{k \cdot z^2}{2} \quad \ldots\ldots \quad [112]$$

62 At either face of the head the intensity s must be zero; i.e., $s = 0$ for $z = \pm\frac{1}{2}t$; therefore, from Eq. [112], $k = -\frac{8s_m}{t^2}$; eliminating k from Eq. [112],

$$s = s_m\left[1 - \frac{4z^2}{t^2}\right] \quad \text{. . . . . [113]}$$

63 Substituting this value for s in Eq. [110],

$$\frac{S}{2\pi x} = s_m \int_{-\frac{t}{2}}^{+\frac{t}{2}}\left[1 - \frac{4z^2}{t^2}\right]dz \equiv \frac{2}{3}s_m \cdot t \quad \text{. . . . . [114]}$$

or

$$s_m = \frac{3s}{4\pi tx}. \quad \text{. . . . . [115]}$$

which is $\frac{3}{2}$ of the mean intensity in the cross-section, and is also the maximum intensity in the cross-section.

64 This analysis of shearing stress does not recognize the presence of bolt holes; but it is nearly correct, for the purpose of finding the maximum intensity of shear in the bolt circle section, to divide the value of S at the bolt circle by the *net* sectional area, and multiply the result by $\frac{3}{2}$.

65 Since $S = m + nx^2$, Eq. [115] may be rewritten as follows:

$$s_m = \frac{3}{4\pi t}\left[\frac{m}{x} + nx\right] \quad \text{. . . . . [116]}$$

66 From general theory, the shear modulus bears the following relation to the modulus of elasticity and Poisson's ratio:

$$\text{Shear modulus} = \frac{E}{2(1+\lambda)}$$

Therefore the angle of detrusion at any point of the neutral surface, or in other words the first derivative of the shear deflection with respect to x, is given by the expression

$$\frac{dY}{dx} = -\frac{S_m}{\frac{E}{2(1+\lambda)}} \quad \text{. . . . . [117]}$$

the negative sign being required by convention.

67 Substituting for s_m according to Eq. [116],

$$\frac{dY}{dx} = -\frac{3(1+\lambda)}{2\pi Et}\left[\frac{m}{x} + nx\right] \quad \text{. . . . . [118]}$$

Integrating,

$$Y = \frac{3(1+\lambda)}{2\pi Et}\left[-\frac{m}{W}\log_e x + D - \frac{n}{2W}\cdot x^2\right] \quad \text{. . . . . [119]}$$

where D is the arbitrary constant of integration. As in the flexure analysis W has been introduced for convenience. Substituting for m and n their values as previously derived, for each of the three zones, we arrive at the following expressions for the shear deflection of the neutral surface:

Inner zone

$$Y = \frac{3(1+\lambda)W}{2\pi Et}\cdot D_1 \quad \text{. . . . . [120]}$$

Loaded zone

$$Y = \frac{3(1+\lambda)W}{2\pi Et}\left[\frac{a^2}{c^2 - a^2}\log_e x + D_2 - \frac{x^2}{2(c^2 - a^2)}\right] \quad \text{. . . [121]}$$

Outer zone

$$Y = \frac{3(1+\lambda)W}{2\pi Et}\left[-\log_e x + D_3\right] \quad \text{. . . . . [122]}$$

68 The three unknown constants D_1, D_2 and D_3 are determined by the following conditions.

69 By definition, $Y = 0$ at the support circle. Hence, by Eq. [**122**],

$$-\log_e R + D_3 = 0 \quad \ldots\ldots\ldots\ldots \quad [\text{I}]$$

70 Y must be continuous across zone boundaries, wherefore Eqs. [**122**] and [**121**] must yield identical values for Y at $x = c$, and Eqs. [**120**] and [**121**] at $x = a$.

$$D_3 - D_2 = \frac{c^2}{c^2 - a^2}\left[\log_e c - \frac{1}{2}\right] \quad \ldots\ldots\ldots\ldots \quad [\text{II}]$$

$$D_1 - D_2 = \frac{a^2}{c^2 - a^2}\left[\log_e a - \frac{1}{2}\right] \quad \ldots\ldots\ldots\ldots \quad [\text{III}]$$

71 Solving these three equations, we find:

$$D_1 = \log_e R - \frac{1}{c^2 - a^2}\left[c^2 \log_e c - a^2 \log_e a\right] + \frac{1}{2} \quad \ldots\ldots\ldots \quad [\mathbf{123}]$$

$$D_2 = \log_e R - \frac{c^2}{c^2 - a^2}\left[\log_e c - \frac{1}{2}\right] \quad \ldots\ldots\ldots\ldots \quad [\mathbf{124}]$$

$$D_3 = \log_e R \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad [\mathbf{125}]$$

72 Eliminating the D's from Eqs. [**120**] to [**122**],

$$\text{Inner zone } Y = \frac{3(1+\lambda)W}{2\pi Et}\left[\log_e R - \frac{1}{c^2 - a^2}(c^2 \log_e c - a^2 \log_e a) + \frac{1}{2}\right] \quad [\mathbf{126}]$$

$$\text{Loaded zone } Y = \frac{3(1+\lambda)W}{2\pi Et}\left[\log_e \frac{R}{x} - \frac{1}{c^2 - a^2}\left(c^2 \log_e \frac{c}{x} - \frac{c^2 - x^2}{2}\right)\right] \quad [\mathbf{127}]$$

$$\text{Outer zone } Y = \frac{3(1+\lambda)W}{2\pi Et} \log_e \frac{R}{x} \quad \ldots\ldots\ldots\ldots\ldots \quad [\mathbf{128}]$$

73 Inasmuch as the distance between the neutral surface and either face remains constant for all values of x and for all conditions of loading (except for the exceedingly small reduction in this distance due to surface application of load, which has nothing to do with detrusion), the values of Y given by these equations apply to the shear deflections of both faces as well as of the neutral surface.

74 With respect to shear, it is not necessary to distinguish between a solid head and one having a central hole.

75 To handle the case of a load concentrated on a circle of radius c, let a approach c as a limit in Eq. [**126**]:

Load concentrated on a circle of radius c:

$$\text{Inner zone} \qquad Y = \frac{3(1+\lambda)W}{2\pi Et} \log_e \frac{R}{c} \quad \ldots\ldots\ldots \quad [\mathbf{129}]$$

Outer zone — Use Eq. [**128**]

76 The treatment of shearing stresses and deflections is thus completed. The formulas are so simple, that it is not thought necessary to discuss their application to special cases.

77 Evidently the total or resultant deflection of the head at any point, is equal to the sum of the resultant flexure deflection y and the resultant shearing deflection Y at that point.

ILLUSTRATIONS

78 Two fairly typical cases have been analyzed for the purpose of illustrating the application of this article, and the results are shown diagrammatically in Figs. 6 and 7. The two cases are alike,

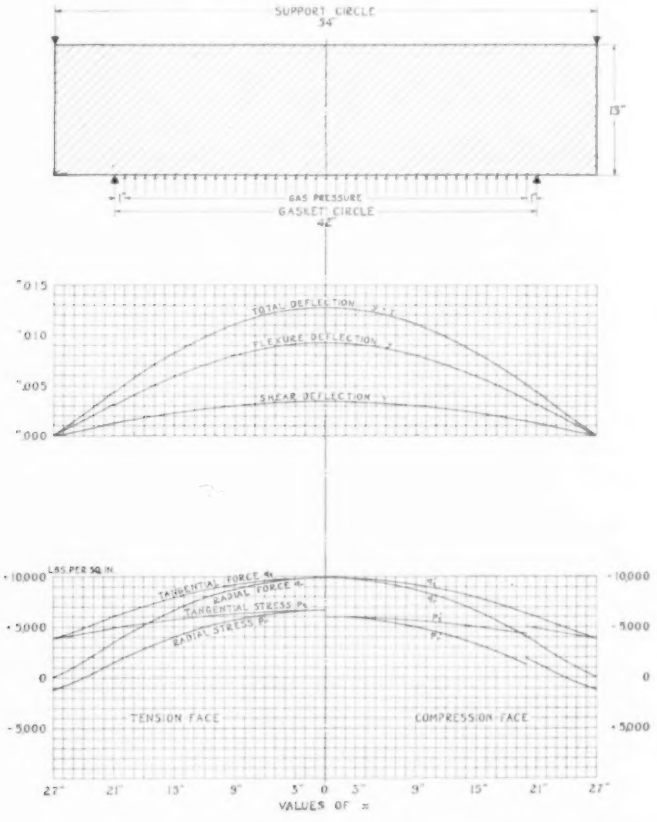


FIG. 6 ANALYSIS OF SOLID CIRCULAR HEAD

except that the first is that of a solid head, whereas the second is that of a head having a central hole.

79 Each head is of steel 13 in. thick, and is simply supported on a circle 54 in. in diameter. The gasket is of appreciable width, but for convenience is considered to form a line contact along a circumference 42 in. in diameter. The total gasket load is 491,300 lb.

Gas pressure is applied over the entire area of a circle 40 in. in diameter, this being the inside diameter of the gasket; the intensity of this pressure is 2000 lb. per sq. in. In the case of the head with the hole, the gas load acting on the stopper of the hole is transmitted to the head as a shear at the edge of the hole, which is 6 in. in diameter.

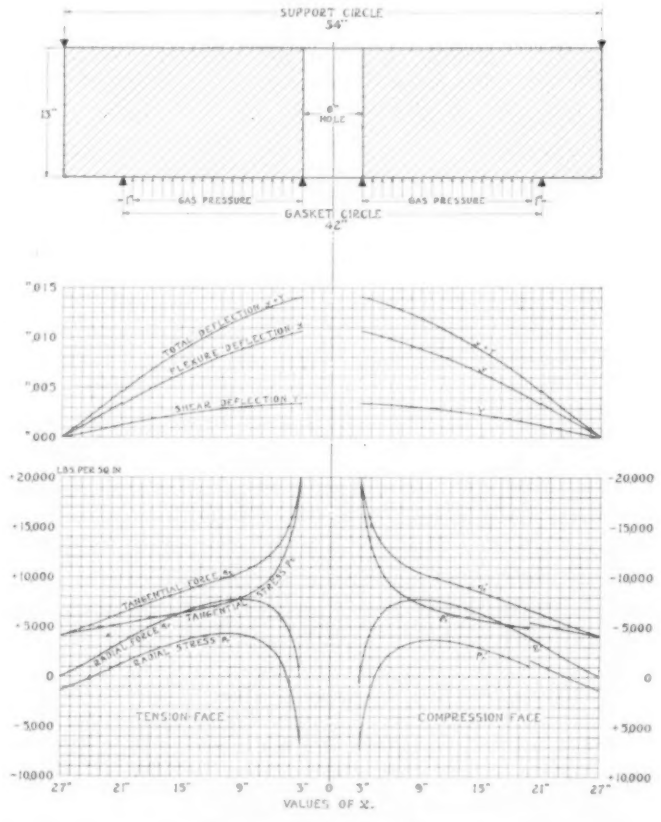


FIG. 7 ANALYSIS OF CIRCULAR HEAD HAVING A CENTRAL HOLE

The steel has a modulus of elasticity of 30,000,000 lb. per sq. in., and Poisson's ratio is taken as $\frac{1}{3}$.

80 The analysis of the solid head is made in two parts, one for the gas load of 2,513,300 lb., and the other for the gasket load of 491,300 lb. For the head with the hole there are three distinct loads: the main gas load, which is 2,456,700 lb. distributed over the

annular space between hole and gasket; the gas load concentrated at the edge of the hole, amounting to the difference between 2,513,300 and 2,456,700, or 56,600 lb.; the gasket load of 491,300 lb.

81 The following are therefore the values of the given constants which apply to all loads in both cases:

$R = 27$ in. $\quad E = 30{,}000{,}000$ lb. per sq. in.
$t = 13$ in. $\quad \lambda = \frac{1}{3}$

82 The following are the values of the other given constants, which are different for different parts of the analyses:

Solid Head	Head with Hole
Gas load — 2 zones, loaded and outer	Main gas load — 2 zones, loaded and outer
$W = 2{,}513{,}300$ lb.	$W = 2{,}456{,}700$ lb.
$c = 20$ in.	$c = 20$ in.
$a = r = 0$	$a = r = 3$ in.
	Gas load on stopper — outer zone only
	$W = 56{,}600$ lb.
	$c = a = r = 3$ in.
Gasket load — 2 zones, inner and outer	Gasket load — 2 zones, inner and outer
$W = 491{,}300$ lb.	$W = 491{,}300$ lb.
$c = a = 21$ in.	$c = a = 21$ in.
$r = 0$	$r = 3$ in.

83 The deflections, forces and stresses are determined separately for the several loads in each case, and the results are added to find the resultant effects. These results are plotted in Figs. 6 and 7. The break in the flexure deflection curve in each case is due to the assumed line application of gasket load, as explained in the analysis, and is numerically equal to 0.000067 in.

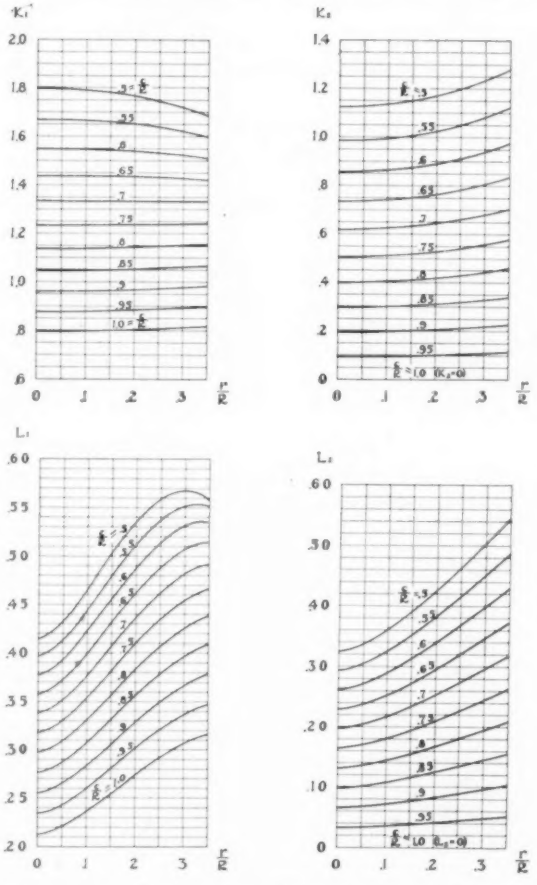
CHARTS

84 Figs. 8 to 15 facilitate the computation of critical stresses and deflections in cases commonly occurring, subject to the assumption of $\frac{1}{3}$ as the value of Poisson's ratio λ[1].

85 Eight auxiliary quantities are introduced, viz. K_1, K_2, L_1, L_2, L'_1, L'_2, M_1, M_2. Each of these, except M_2, is a function of r/R and c/R, r/R being zero in the case of a solid head; M_2 is a function of c/R alone. Each of the eight charts yields the value of one of these auxiliaries in any given case.

[1] For steel, $\frac{1}{3}$ is a convenient approximate value commonly employed; scarcity of experimental information in regard to lateral strains makes it useless to seek refinement in regard to the value of λ.

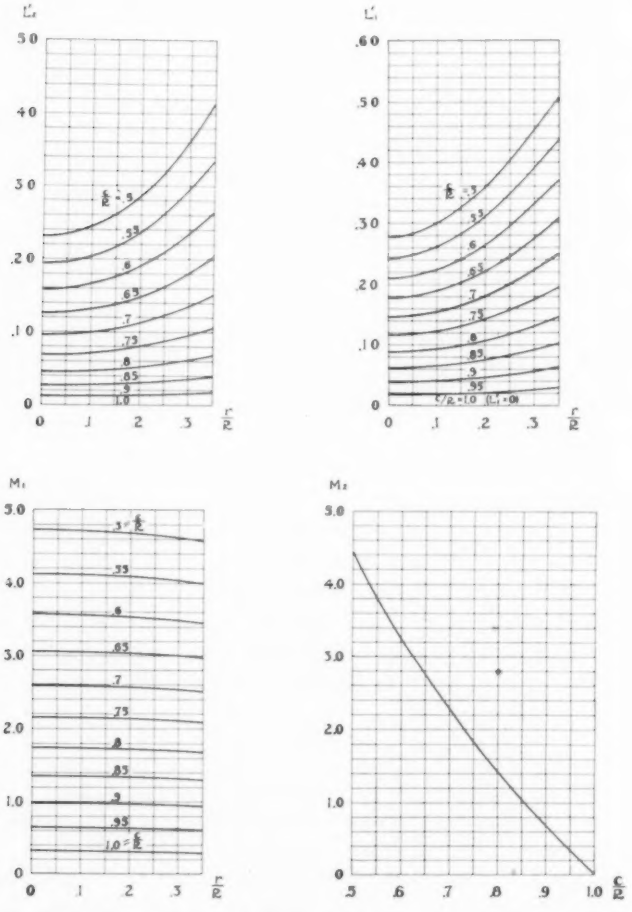
86 The simple formulas classified in Table 1 give the stresses and the deflections at the center in the case of a solid head, or the tangential stresses and the deflections at the edge of the hole in the case of a centrally bored head, due to any load W uniformly distributed



FIGS. 8–11 CHARTS FOR FACILITATING COMPUTATION OF CRITICAL STRESSES AND DEFLECTIONS IN COMMONLY OCCURRING CASES, POISSON'S RATIO BEING TAKEN AS $\frac{1}{3}$.

over the entire area of a circle of radius c concentric with the support circle, or due to any load W concentrated along the circumference of such a circle. These formulas give also the deflections at the circumference of radius c.

87 The formulas giving the effects of a distributed load, in the case of a head with a central hole, include the effects of that portion of the load which acts on the area of the hole and which



FIGS. 12–15 CHARTS FOR FACILITATING COMPUTATION OF CRITICAL STRESSES AND DEFLECTIONS IN COMMONLY OCCURRING CASES, POISSON'S RATIO BEING TAKEN AS $\frac{1}{3}$

is transmitted to the edge of the hole by the plug, cap, pipe or other stopper which closes the hole; the proper value to use for W is the total pressure within the circle of radius c.

88 The formulas by means of which the auxiliary quantities were computed, are not reproduced here, being cumbersome; they were reduced directly from formulas derived in the earlier part of this article, with λ put equal to $\frac{1}{3}$. It is to be noted that these auxiliaries do not serve to give stresses and deflections *at all points*, as do

TABLE 1 AUXILIARY FORMULAS FOR USE WITH CHARTS, FIGS. 8 TO 15

	Symbol	Solid Head (put $r/R = 0$)	
		Due to load W distributed over area of circle, radius c	Due to load W concentrated on circumference, radius c
Intensity of stress in tension face at center	$p_t = p_r$	$\frac{W \cdot K_1}{3t^2}$	$\frac{W \cdot K_2}{3t^2}$
Intensity of stress in compression face at center	$p'_t = p'_r$	$-\frac{W}{3}\left(\frac{K_1}{t^2} - \frac{1}{\pi c^2}\right)$	$-\frac{W \cdot K_2}{3t^2}$
Center deflection due to flexure	y	$\frac{W}{Et}\left(\frac{R^2 L_1}{t^2} - \frac{1}{6\pi}\right)$	$\frac{W}{Et}\left(\frac{R^2 L_2}{t^2} - \frac{1}{6\pi}\right)$
Center deflection due to shear	Y	$\frac{WM_1}{E \cdot t}$	$\frac{WM_2}{Et}$
Deflection at radius c due to flexure	y	$\frac{WR^2 L'_1}{Et^3}$	$\frac{W}{Et}\left(\frac{R^2 L'_2}{t^2} - \frac{1}{12\pi}\right)$
Deflection at radius c due to shear	Y	$\frac{WM_2}{Et}$	$\frac{WM_2}{Et}$
		Head with Central Hole of Radius r	
	Symbol	Due to load W distributed over area of circle, radius c	Due to load W concentrated on circumference, radius c
Intensity of tangential stress in tension face at edge of hole	p_t	$\frac{W \cdot K_1}{t^2}$	$\frac{W \cdot K_2}{t^2}$
Intensity of tangential stress in compression face, edge of hole	p'_t	$-W\left(\frac{K_1}{t^2} - \frac{1}{3\pi c^2}\right)$	$-\frac{W \cdot K_2}{t^2}$
Deflection at edge of hole due to flexure	y	$\frac{W}{Et}\left(\frac{R^2 L_1}{t^2} - \frac{1 - r^2/c^2}{6\pi}\right)$	$\frac{W}{Et}\left(\frac{R^2 L_2}{t^2} - \frac{1}{6\pi}\right)$
Deflection at edge of hole due to shear	Y	$\frac{WM_1}{Et}$	$\frac{W \cdot M_2}{Et}$
Deflection at radius c due to flexure	y	$\frac{WR^2 L'_1}{Et^3}$	$\frac{W}{E \cdot t}\left(\frac{R^2 \cdot L'_2}{t^3} - \frac{1}{12\pi}\right)$
Deflection at radius c due to shear	Y	$\frac{WM_2}{Et}$	$\frac{WM_2}{Et}$

the formulas of the analysis; they give only particular stresses and deflections of major importance.

89 The main practical purpose served by the charts is as follows. In the case of a steel cylinder head of uniform thickness t, simply supported around a circle of radius R, having a central hole of radius r (which is considered zero in case there is actually no such

hole), sustaining a gasket load concentrated along a circumference of radius c, and sustaining also a static fluid pressure acting on the entire area enclosed by the gasket circle, *to find the maximum intensities of tensile and compressive stress, the maximum deflections due to flexure and shear, respectively, and the deflections at the gasket circle due to flexure and shear, respectively.*

DISCUSSION

W. H. RUNKEL, in opening the discussion, asked what factor of safety was used in these flat cylinder heads designed for the nitrate plant.

THE AUTHOR replied that a maximum intensity of direct stress of 10,000 lb. per sq. in. was allowed. The material was forged steel, and was expected to be a special alloy. Tests were being made at the Bureau of Standards, at the time the author severed his connection with the Bureau, to determine its strength at high temperature, and the effect of a hole in it. Many specimens composed of carbon steel, nickel-carbon steel, chrome-vanadium steel, and molybdenum steel were being pulled at high temperatures. The ultimate strength of all the alloys was increased by elevating the temperature to 300 or 400 deg. fahr., but there was a tendency to decline beyond these values, which finally dropped off decidedly at more elevated temperatures.

CONRAD C. JACOBSON asked if there was not a limit to the efficient thickness of the head, where a very high pressure was developed and there was a tendency toward bending between the bolt circle and the gasket circle. Would it not be a better design, he asked, to use hydraulic clamps directly through the gasket circle, rather than to depend on the bolt circle, and thus do away with this flexure.

THE AUTHOR answered that actual trials with screwed-in heads with the thrust directly against the gasket resulted in serious leakage.

CONRAD C. JACOBSON said that this was because no follow-up had been provided and when the elasticity was gone, there was a space between the head and the flange which decreased the initial tension on the gasket. He said that he was talking from experience, in work in which old-fashioned methods were used, but trouble was

found in keeping tight joints. No matter how thick the heads were made, there was an initial pressure on the gasket with bolts.

THE AUTHOR said that his method was to calculate the deflation or bulging between the bolt circle and the gasket under a condition of bolt tension, without gas pressure, and also under the condition of bolt and gas pressure. If the overhang on the head beyond the gasket was properly proportioned to the other elements, there was a residual gasket pressure after the gas pressure was applied, and it was found feasible to do with a residual gasket pressure equal to about one-fifth of the total gas pressure. The elasticity of the plate made it possible to retain a residual gasket pressure after the gas pressure was applied.

CONRAD C. JACOBSON asked if it was always possible to determine the exact pressure that would be developed by the liquid pressure of the material handled.

THE AUTHOR answered that exact calculations were theoretically but not actually possible. The pressure in the system was maintained by means of a standard type of compressor, and the loss of head through the various members of the circulating system was quite definitely known, so that operating conditions were subject to fairly accurate control, both as to pressure in the apparatus and as to temperature.

In dealing with pressure inside any containers used for other purposes than the ones he was dealing with, he would expect that the calculated pressures would be subject to variations due to impurities of catalytic agents, and, that certain mechanical thermal conditions would affect the chemical equilibrium between the reagents and their product which could not be calculated exactly. Therefore, unless mechanical means were used for maintaining the system at a fixed pressure, a considerable percentage of variation in the pressure might be expected.

In connection with his problem, there were incidental temperatures to deal with, which meant that any construction of head or gasket which did not permit relative expansion laterally between the head and gasket seat, would fail.

The bolting of the flanges seemed the most practical method for holding the gas, but even that was not very simple, because wrenches six feet long had to be used to turn up the bolts, and the ends of the wrenches had to be hit with sledges.

CONRAD C. JACOBSON, the author and Thomas W. Milner discussed methods of fastening cylinder heads with clamps.

W. J. DRISKO[1] said that he had been interested in photoelasticity in connection with a number of problems of engineering interest, and the possibility of studying the distribution of stresses in a disc with a pneumatic pressure applied over the surface had occurred to him. He wondered if any information had been obtained through such means as this on the subject of the paper.

THE AUTHOR knew of no experimental evidence which would help Mr. Drisko. Strains, as put forth in his paper, have been completely analyzed. The study of the inside of a thick plate, subject to flexure, showed that the normal elements which were straight before flexure, became curved after flexure, not due to that portion of the deflection which is called flexure but due to detrusion or shear. This was so slight that it would take a photo-micrometer to detect it.

W. J DRISKO said that he had noticed in boring presses that the stress at the edge of the hole is three times the main stress.

THE AUTHOR answered that the tangential direct stress around the edge of a small hole at the center of the disc is approximately, by all the theory, three times the corresponding stress in the case of a solid head having no holes. The reason is that the supporting effect of the compression or tensile stress at right angles to the tangential stress is removed by the hole, thereby removing the reducing effect. This is equal to 1 minus S, or $\frac{2}{3}$, according to Poisson's ratio. The flexure is confined to one direction at this point. These two things sound nearly alike, but they are really distinctly different.

CONRAD C. JACOBSON asked if the size of the hole would have any effect.

THE AUTHOR replied that it does have an effect, but it is possible to analyze it as has been done in the paper. For the stress caused in the case of a pinhole the deflection at the center has as its limits the deflection at the center of the solid head, but the direct stress of tension or compression at the edge of the hole is three times, when Poisson's ratio is the third, what it is in the solid head. In other words, at the instant when the radius of the hole becomes zero, the stress suddenly drops.

[1] Mass. Inst. of Technology, Cambridge, Mass.

If the hole is large, and larger than an infinitesimal hole, the equations yield different results, and so far as the varying stress is concerned, the size of the hole has to become quite large before it makes a very great difference.

In other words, so far as the thickness of head required for safety is concerned, it makes very little difference in a solid head, say, five feet in diameter, whether the hole is for a one-inch pipe, or whether it is a six-inch hole. But there is an enormous difference between the solid head and the one with the hole on the sides, due to the intense stress at the immediate vicinity of the hole only. The boss around the hole will reinforce it, but in the case of a forging, the hole is exceedingly expensive, because the forging is usually made as a disc machined down, so that just as much metal is used for a plate which is ten inches thick with a three-inch thick boss, as for a 13-inch plate.

CONRAD C. JACOBSON asked if the author had ever made a mathematical analysis of a plate uniformly loaded on one side and supported on three or more points on the other side.

THE AUTHOR replied that he did not know whether a rigorous analysis was possible. He believed, however, that close approximations could be made.

ELWYN E. SEELYE. The method of analysis used by the author is based upon the elastic theory and is one of the most interesting examples of applied mathematics of which the writer is familiar. Where homogeneous material is in question if the correct equations are written very exact results may be looked for.

The writing and solving of these differential equations required a mind with considerable aptitude for and knowledge of higher mathematics.

It is interesting to examine the difference in results obtained by analyzing a cylinder head by ordinary statics as compared to the far more correct analysis based on the elastic theory.

In order to make the case as simple as possible let us take the case of a circular plate uniformly loaded and simply supported around its perimeter, taking the moment about a diametrical section. The moment is readily obtained by determining the location of the center of gravity of the semi-circle and the half perimeter respectively. This gives us a total moment through the diametrical section

of $\frac{\omega r}{3\pi}$ which gives an average unit stress of $\frac{\omega}{\pi l^2}$ which if assumed to be parabolic in distribution gives a maximum stress at center of $\frac{1.5\omega}{\pi l^2}$ as compared with $\frac{0.83\omega}{\pi l^2}$ by elastic formula.

This contrast is even greater if there is a small hole in the center of the head when the maximum stress is tangential and is three times that of a solid head, whereas the statical solution would increase the stress only directly with the material omitted.

These surprising differences are met with by engineers in floor construction designs particularly in the case of flat slab construction and affect the mechanical engineer in relation to the strength and economy of his plant housing structures.

No. 1806

PORTING AND CHARGING OF TWO-STROKE OIL ENGINES

By Louis Illmer, Lansdowne, Pa.

Member of the Society

This paper deals primarily with single-acting semi-Diesel engines having the lower portion of the cylinder bore provided with intake and exhaust ports which are successively uncovered by the power piston as it approaches the end of its expansion stroke. In such engines the under side of the piston is generally made to serve as a low-pressure base chamber or crankcase pump for slightly precompressing the air supply and charging the power cylinder.

The advantages of this simple and effective method of charging the power cylinder are obvious, and the author undertakes to present a rational basis for predetermining the correct timing and porting of such valves. A considerable number of semi-Diesel two-stroke-cycle engines have been examined for port characteristics and some of the more promising designs have been critically analyzed to ascertain the reason for their superior performance. The results of this research disclose the particular combination of port relations and corresponding base pressures that are likely to give the best performance under any specific service conditions.

The various losses in fuel economy encountered in direct-injection oil engines are traced in some detail, and a new method is presented for estimating the expected power capacity of a given cylinder design as based upon the percentage of air that may be brought to the support of the injected fuel. The need of keeping all of the air charge in close proximinity with the spray valve and the importance of turbulence in effecting an even distribution of finely atomized fuel is also discussed. In closing, the author points out the line along which the further development of direct-injection or semi-Diesel engines should be looked for.

THE two-stroke-cycle oil engine, because of its simplicity of construction, is finding a wide and rapidly expanding market, particularly as applied to the so-called semi-Diesel or solid-injection engine. As a type, however, it is still in a state of transition and the aim of the present paper is to stimulate further interest in carrying the technique of its design to a higher state of perfection.

2 Primarily, the treatment will cover single-acting semi-Diesel engines having the lower portion of the cylinder bore provided

Presented at the Annual Meeting, December 1921, of The American Society of Mechanical Engineers.

with intake and exhaust ports which are successively uncovered by the power piston as it approaches the end of its expansion stroke. In such engines the under side of the piston is generally made to serve as a low-pressure base chamber or crankcase pump for slightly precompressing the air supply and charging the power cylinder.

3 Piston-controlled ports, as used in the two-cycle engine, eliminate the need of mechanically operated inlet and exhaust valves, and their design has long been a fascinating subject for invention on the part of engineers. The advantages of this simple and effective method of charging the power cylinder are obvious, and the author undertakes to present a rational basis for predetermining the correct timing and porting of such valves.

4 A considerable number of semi-Diesel two-stroke-cycle engines have been examined for port characteristics and some of the more promising designs have been critically analyzed to ascertain the reason for their superior performance. The results of this research disclose the particular combination of port relations and corresponding base pressures that are likely to give the best performance under any specified service conditions.

5 In two-stroke-cycle engines the products of combustion remaining within the cylinder from the preceding burnt charge are swept out by means of a fresh charge of incoming air. The process being a difficult one, correct technique is required to displace the products of combustion effectively. The more perfectly this is accomplished, the larger will be the pure-air content held within the available cylinder volume at the instant of exhaust closure. The air so trapped is a direct measure of the amount of fuel oil that can be efficiently burned during the succeeding working stroke.

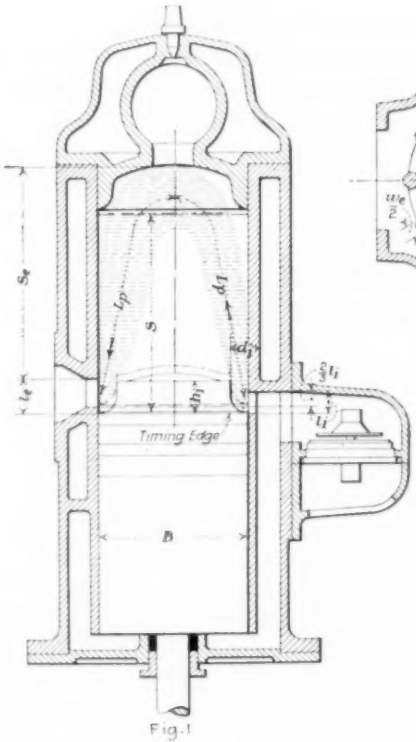
6 The aim in an engine of this type is to charge the maximum of fresh air into the power cylinder at the minimum expenditure of pump work. The designer is therefore obliged to work with a relatively small amount of scavenging air and to depend upon its skilful introduction as a means of insuring an unpolluted working charge.

PISTON DEFLECTOR PLATES

7 One of the most effective means for properly directing the air charge into a two-stroke-cycle cylinder provided with piston controlled intake ports is the use of a baffle plate cast upon the piston head. This deflector lug may readily be made to coact with the intake port in such fashion that the incoming air charge is sharply deflected up into the cylinder bore. The resulting long sweep given

to the path of flow facilitates the spreading of the incoming stream line and intercepts the direct flow of air toward the open exhaust ports.

8 Furthermore, this method sets up intense eddy currents both at the intake-port edges and at the point where the deflector



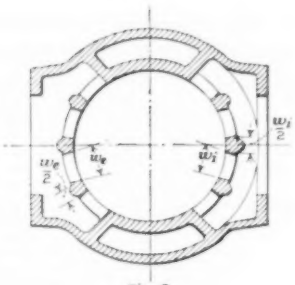


FIG. 1 TWO-STROKE-CYCLE POWER CYLINDER
FIG. 2 SECTION THROUGH CYLINDER PORTS

lug interferes with the normal course of air flow. These eddy losses are capable of absorbing the major portion of the flow energy and of abruptly lowering the high velocity of the incoming air immediately after passing the nozzle zone.

9 Such absorption of flow energy in close proximity to the intake nozzle confines the incoming charge to a body-like formation and avoids excessive intermingling with the displaced products of combustion. In case the charge is not properly deflected and made

to spread out on its way through the cylinder, the incoming air tends to head directly toward the open exhaust port without displacing the hot burnt gases lying in the upper portion of the bore. The resulting deficiency in combustion air leads to low power capacity per unit of piston displacement.

BASE CHARGING PRESSURES

10 The average pressure head required to inject the air charge into the power cylinder depends upon three principal factors:

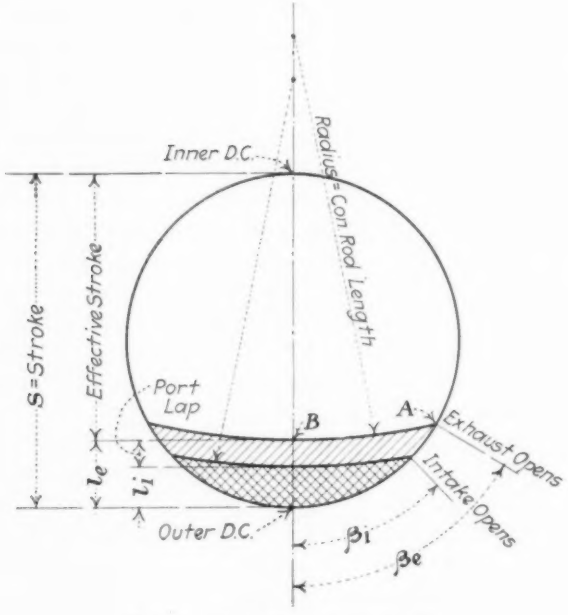


FIG. 3 PORT-TIMING DIAGRAM

a The volume of precompressed air delivered to the intake port from the base chamber

b The average inlet-port area exposed by the piston during the base delivery period

c The time available for introducing the charge into the power cylinder.

The average back pressure against which the discharge from the base chamber must work is largely fixed by the area and lead given to the exhaust port. The intake port should not be allowed to open

until the exhaust back pressure has dropped to a value about equal to the maximum pressure in the base chamber. The various factors that control the rate of exhaust drop will be discussed after taking up the base-pressure head required to inject a given cylinder charge against a negligible back pressure.

INTAKE-PORT AREAS

11 Fig. 1 shows a typical sectional assembly of a semi-Diesel two-stroke-cycle cylinder. A sectional view taken through the cylinder ports is shown in Fig. 2, while the timing of the intake and the exhaust ports is diagrammatically indicated in Fig. 3. The following constants will be required to characterize the porting relations:

B = cylinder bore, in.
S = piston stroke, in.
D = piston displacement, cu. in.
$r = S/B$ = stroke-to-bore ratio
A_p = piston area, sq. in.
β_i, β_e = angular intake and exhaust port timing as respectively fixed by port dimensions, deg. (see Fig. 3)
$l_i = S/C_i$ = linear intake-port timing, in. (see Figs. 1 and 3)
$l_e = S/C_e$ = linear exhaust-port timing, in.
C_i, C_e = characteristic port constants for the intake and the exhaust port
n = number of component intake or exhaust ports, each having a width w_i or w_e (see Fig. 2)
$W_i n/\pi B = X_i$ = ratio of perimetric intake-port length to bore circumference
$W_e n/\pi B = X_e$ = ratio of perimetric exhaust-port length to bore circumference
$a_i = \frac{2}{3}(nw_i l_i)$ = average intake-port area exposed by the piston during the crank movement β_i, sq. in.

12 Fig. 3 shows the angular position β_e assumed by the crank at the instant of exhaust-port opening. The corresponding position of the piston may be located by dropping from the point A to B along an arc whose radius is equal to the length of the connecting rod. As indicated, the exhaust port opens at a distance l_e from the outer dead-center position of the piston. The angular crank position β_i for the intake-port timing bears a similar relation to the length l_i.

13 As shown in Fig. 1, the timing edge of a two-stroke-cycle piston is usually made to align with the bottom of the intake port when the piston reaches its outer dead-center position. Assuming the time for introducing the entire base discharge into the power cylinder to be exactly equal to the crank-angle period β_i, then the average intake-port area available for transferring the charge may be approximated by working with the mean port-opening position $\frac{2}{3}l_i$ of the piston, as indicated by dotted lines in Fig. 1.

14 The accompanying coefficient of discharge for piston-controlled intake ports may be taken at a constant value of $\frac{2}{3}$. Accordingly, the average contracted-nozzle area uncovered by the piston between the instant of opening and the time of reaching its outer dead-center position becomes equal to

$$A_i = \tfrac{2}{3} \cdot \tfrac{2}{3} l_i(X_i \pi B) = \frac{1.4 r X_i B^2}{C_i} \quad . \quad . \quad . \quad . \quad . \quad . \quad [1]$$

where $A_i = \frac{2}{3}a_i$ = average contracted nozzle area of the intake port exposed by the piston up to its dead-center position, sq. in.

Taken in terms of piston area,

$$\frac{A_i}{A_p} = 1.78 \frac{r X_i}{C_i} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [2]$$

It will be seen that the intake-nozzle area bears a fixed ratio to the piston area for given values of r, X_i and C_i, all of which are usually kept in a fairly constant relation for any given line of engines. The ratio X_i, representing the relation between the intake-port length and the bore circumference, should be made reasonably large. Its value ranges between $\frac{1}{5}$ and $\frac{1}{3}$, with $\frac{1}{4}$ as a good basic value.

15 The given area A_i applies only in case the completion of the charging period occurs at the precise instant that the piston reaches its dead-center position. This condition is satisfied when the pressure head is just sufficient to drive the entire base-chamber discharge through the given area by the time the piston has fully exposed the intake port, all of which is indicated in the "nominal" diagram Fig. 5, and will be discussed later.

16 When a pressure higher than the nominal is used, the base chamber will vent itself into the power cylinder in less time than corresponds to the crank angle β_i and the piston will therefore be unable to utilize the full average contracted-nozzle area A_i. A numerical value for the average intake area that will be exposed by the piston under this partially open condition may be deduced as follows:

17 The angle α_i in Fig. 4 represents the drop time required for the base pressure to vent itself into the cylinder, assuming the corresponding partial port length l_i' exposed at the end of the charging period to be less than the full length l_i. If the slope of the line z-z for the angle $\left(\beta_i - \frac{\alpha_i}{2}\right)$ be taken as a basis for fixing the relation between the length l_i' and the drop angle α_i, the following empirical formula will be found approximately correct when using a connecting-rod length of about $2S$:

$$\alpha_i C_i' = \frac{8500}{\beta_i - \frac{\alpha_i}{2}} \quad \ldots\ldots\ldots\ldots [3]$$

where α_i = angular drop time (less than β_i) required for the base pressure to completely charge the power cylinder, deg.

$C_i' = S/l_i'$ = constant for the partial intake port length l_i' exposed by the piston during the crank angle α_i.

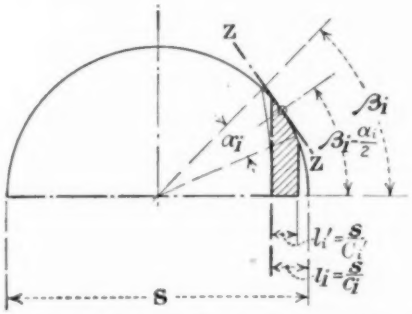


FIG. 4 DIAGRAM SHOWING RELATION BETWEEN α_i AND l_i

When the angle α_i becomes equal to β_i, the intake port will be entirely exposed by the time the cylinder has been fully charged, and the above relations simplify to

$$C_i \beta_i^2 = 17{,}000 \quad \ldots\ldots\ldots\ldots [4]$$

BASE-CHAMBER PRESSURES

18 The term "non-scavenging" will be applied to two-stroke-cycle engines in which the power piston alone furnishes all the displacement imparted to the base or crank chamber. When this limited displacement is supplemented by additional pumping means, the designation "scavenging engine" will be used.

19 The amount of free air actually discharged from the base chamber of a non-scavenging two-stroke-cycle engine generally lies between $\frac{3}{4}$ to $\frac{7}{8}$ of the power-piston displacement. For normal proportions of valve area and base pressure the volumetric efficiency of the base-chamber discharge may be conservatively placed at $0.8D$, and it will be so used in the following discussion.

20 A nominal base or crankcase diagram for a non-scavenging engine is shown in Fig. 5. The maximum pressure P_i is reached shortly after the intake port opens. The counter pressure that must be overcome by the incoming charge is partly fixed by the rapidity with which the exhaust pressure equalizes with the atmosphere. When the exhaust release is given ample lead as indicated, the back pressure will drop well below the maximum base pressure prior to intake opening. In that event practically the full average

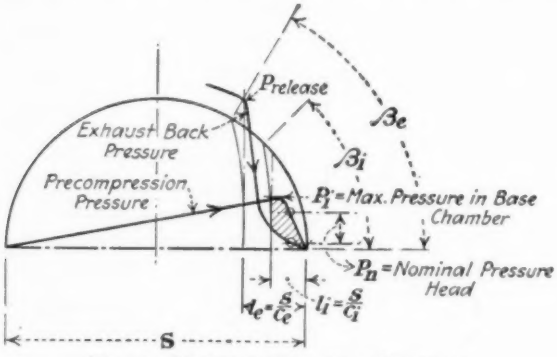


FIG. 5 NOMINAL CHARGING DIAGRAM

base-pressure head acting behind the intake port becomes available to drive the charge into the power cylinder.

21 The effective average pressure head which is available for driving the charge out of the base chamber and through the intake port should be measured by the mean ordinate P_n of the cross-sectioned area shown in Fig. 5, which then represents the net or effective head maintained by the base chamber in charging the power cylinder up to the time the piston reaches its dead-center position.

NOMINAL PRESSURE HEAD

22 The simplest method of arriving at the mean pressure head required to charge a non-scavenging two-stroke-cycle engine consists in first fixing upon a nominal or arbitrary reference pressure

head P_n applicable to certain ideal charging conditions. The actual pressure head P_{hd} that may be required under any modified conditions, such as a change in the charging time, etc., can most readily be determined by allowing for the difference in intake nozzle velocity as taken with respect to such a standard.

23 The nominal pressure head P_n will be assumed capable of forcing the full normal base discharge of a non-scavenging engine through the average nozzle area A_i against a negligible back pressure by the time the piston reaches its dead-center position. When passing the stipulated base discharge of $0.8D$ through a contracted

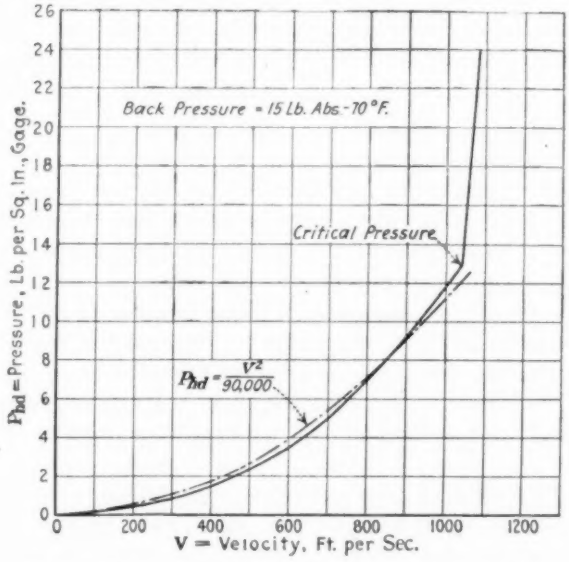


FIG. 6 THEORETICAL NOZZLE VELOCITIES FOR AIR

intake area A_i in the charging time β_i, the resulting average nozzle velocity becomes equal to:

$$V_n = 1.35\frac{(\text{P.S.})}{X_i r} \cdot \frac{C_i}{\beta_i} = 23{,}000\,\frac{(\text{P.S.})}{X_i r \beta_i^3} \quad . \quad . \quad . \quad . \quad . \quad [5]$$

where V_n = nominal velocity required in the intake nozzle to charge a non-scavenging two-stroke engine in the period β_i, ft. per sec.

N = revolutions per minute

$(\text{P.S.}) = \frac{SN}{6}$ = piston speed, ft. per min.

24 The average pressure head required to set up the velocity V_n is shown in Fig. 6, which represents the theoretical nozzle velocities for air flowing against atmospheric back pressure as determined from thermodynamic formulas.[1] Below the critical pressure of about 13 lb. gage this curve may be closely approximated by the relation:

$$P_{hd} = \frac{V^2}{90{,}000} \text{ or } \sqrt{P_{hd}} = \frac{V}{300} \quad \ldots\ldots\ldots [6]$$

where P_{hd} = pressure head, lb. per sq. in. gage, required to set up a maximum air nozzle velocity of V ft. per sec.

25 The desired value for the average pressure head P_n required to produce the nominal nozzle velocity V_n may now be found by substituting into Equation [6], thus

$$\sqrt{P_n} = 75\frac{(\text{P.S.})}{rX_i} \times \frac{1}{\beta_i^3} \quad \ldots\ldots\ldots [7]$$

where P_n = nominal pressure head required to set up a nominal nozzle velocity V_n for an effective base discharge of $0.8D$ when fully injecting the charge of a non-scavenging engine during the crank-angle period β_i.

The base pressure head P_n varies as the square of the piston speed, but as is evident from Equation [7], the dominating factor in fixing this head is the intake timing angle β_i. As derived from this equation, the nominal head P_n applying to normal port characteristics is plotted in Fig. 7. These curves show that P_n increases rapidly for any angle β_i less than 50 deg., especially for small stroke-to-bore ratios.

26 The nominal pressure head given by Equation [7] is intended to serve as a reference standard, and the actual pressure that may be required under any other conditions, such as a change in the charging period or an increase in the amount of base discharge, may be allowed for by introducing a suitable constant to compensate for the velocity difference that must be set up through the intake nozzle under such modified conditions.

RATE OF EXHAUST DROP

27 The maximum base pressure P_i required to produce a given effective pressure head P_{nd} is contingent upon the rate of exhaust back-pressure drop. It is therefore necessary to fix upon the exhaust drop line before a numerical value for the maximum base pressure can be arrived at.

[1] Based upon Zeuner's Technische Thermodynamik.

28 Referring again to Fig. 6, showing the nozzle velocities corresponding to different pressure heads, it will be noted that the law of flow changes abruptly above the critical pressure of about 13 lb. gage. At this point the velocity in the nozzle has become equal to the velocity of sound in atmospheric air, and this critical velocity cannot be further increased except by a change of density in the air egressing from the nozzle.

29 The major portion of the released exhaust gases escape from the cylinder in accordance with this law of flow. For the sake

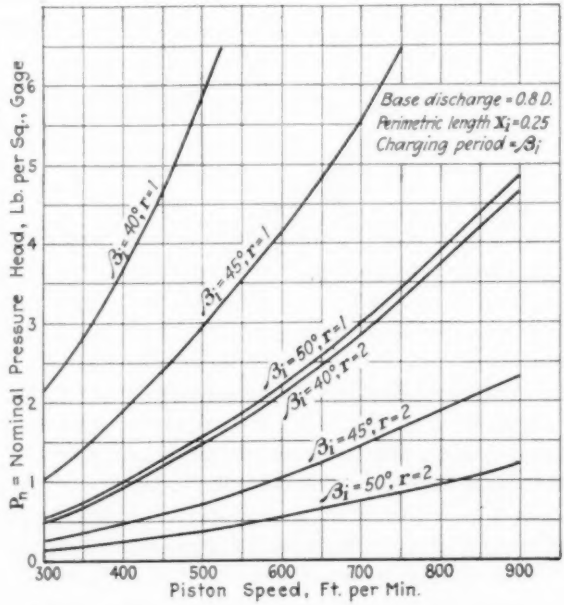


FIG. 7 NOMINAL PRESSURE HEAD VS. PISTON SPEED

of simplifying the determination of the time of drop, it may be assumed that the entire volume of exhaust gas is discharged at a certain uniform sound velocity in the time t_α, which may be designated as the "straight line" drop time as indicated in Fig. 8 by the line *CD*.

30 As determined from a series of light-spring indicator diagrams, the retarded rate of exhaust drop, which occurs after the release pressure falls below the given critical pressure of 13 lb., may

be allowed for by taking the total drop time at $2t_\alpha$, i.e., about twice that required for the straight-line period. The intermediate points on the back-pressure curve may be approximated by means of the graphical construction for a hyperbolic expansion line. By dividing the ordinates into any number of equal parts as indicated in Fig. 8, the desired curve will be outlined by the tangents 1–1, 2–2, etc., and this may be used to fix the exhaust back pressure against which the base pressure must work during the cylinder-charging period.

31 In accordance with Equation [3], the average exhaust vent

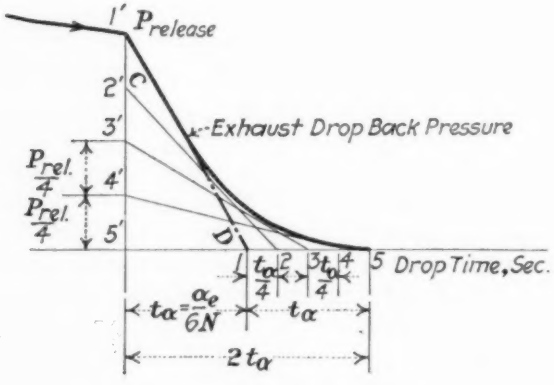


FIG. 8 EXHAUST BACK-PRESSURE LINE

area exposed by the piston during the straight-line drop time t_α may be taken as a function of the angle β_e, thus:

$$a_e C'_e = \frac{8500}{\beta_e - \frac{a_e}{2}} \quad \ldots\ldots\ldots\ldots [8]$$

where a_e = angular straight-line drop angle required for the released full-load exhaust pressure to equalize with the atmosphere when discharging at the velocity of sound through the average nozzle area exposed by the piston during the time t_α, in degrees.

The principal factors controlling the rate of drop through the exposed exhaust port as fixed by the angle a_e are not unlike those previously deduced for the rate of air discharge from the base chamber. Owing, however, to the uncertainty in the assumptions that must be made in the case of the exhaust drop, it is advisable to

work with empirical constants as determined from actual indicator cards.

32 Without elaborating on its derivation, the following equation may be used for finding the expected rate of pressure drop over a piston-controlled exhaust port when releasing from about 30 to 35 lb. gage against normal back-pressure conditions:

$$\alpha_e \sqrt{\beta_e - \frac{\alpha_e}{2}} = 3\sqrt{\frac{(\text{P.S.})}{rX_e}} \quad \ldots\ldots\ldots [9]$$

33 In using this formula it is necessary to assume a preliminary value for the drop angle α_e and then interpolate until the given relation is satisfied. For good oil-engine practice this angle usually lies between the values of $\frac{1}{2}\beta_e$ and $\frac{1}{3}\beta_e$, and for present purposes it may be set at $\frac{3}{8}\beta_e$. Accordingly, the exhaust-port timing angle should not be made less than

$$\beta_e = 4.5\sqrt[3]{\frac{(\text{P.S.})}{rX_e}} \quad \ldots\ldots\ldots [10]$$

34 The rate of exhaust drop is only remotely dependent upon piston speed, as is evident from Fig. 9, which shows the required exhaust-port timing as determined from this equation on the basis of $X_e = \frac{1}{4}$. These curves indicate that for short-stroke high-speed oil engines of the two-stroke-cycle type the angle β_e should be made equal to about 60 deg., while for long-stroke slow-speed engines the exhaust lead may be reduced by approximately 10 deg.

35 The exhaust back pressure as determined for full rated speed should, if possible, be made to drop well below that existing in the base chamber at the instant of intake-port opening. If at this critical period the exhaust pressure is allowed to exceed $\frac{5}{4}$ to $\frac{3}{2}$ of the base pressure, a blow-back is to be expected, and this will seriously interfere with the proper charging of the power cylinder.

STANDARD CHARGING DIAGRAM

36 While the nominal base diagram shown in Fig. 5 assumes the power cylinder to be fully charged by the time the piston reaches its dead-center position, it is customary in practice to extend the actual charging time somewhat, and in the case of the commercial non-scavenging oil engine only about two-thirds of the base discharge is usually released during the specified β_i period when running at full rated speed.

37 For any given port relations, such prolongation of the charging period correspondingly reduces the required nozzle velocity and allows the nominal pressure head P_n, as fixed by Equation [7], to be lowered to about $P_{hd} = P_n/2$. By the time the piston reaches its outer dead-center position, the base pressure will only have dropped to about $\frac{1}{3}P_i$, but this is still sufficient to drive the remaining charge from the base chamber prior to intake-port closure. At slower speeds

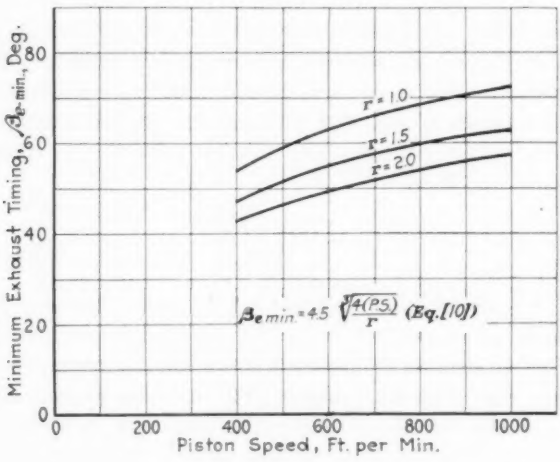


FIG. 9 ANGULAR EXHAUST-PORT TIMING

the reduced exhaust drop angle α_e allows the base pressure to vent itself more rapidly than at full speeds.

38 A full-speed diagram, applying to a commercial non-scavenging oil engine, is shown in Fig. 10, and it will be designated as the "standard" charging diagram. Under the given conditions it is necessary to maintain an average pressure head of about $P_n/2$ with respect to the full-speed exhaust back pressure, as indicated by the average vertical ordinate of the cross-sectioned portion of the diagram area.

39 The factors upon which the maximum base pressure depends may be deduced on the following basis: The effective pressure head $P_n/2$, as found from Fig. 10, usually bears a more or less fixed ratio to the maximum base pressure P_i, and, in the case of the standard diagram, this may be expected to lie between the values of $P_i/3$ and

$P_i/2$. Taking an average value at $\frac{1}{2}P_n = \frac{3}{8}P_i$, and limiting P_i to the critical pressure of 13 lb. gage, the maximum base pressure becomes equal to about

$$P_i = \tfrac{4}{3}P_n = 7500\left\{\frac{(\text{P.S.})}{rX_i\beta_i^3}\right\}^2 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [11]$$

where P_i = maximum base pressure required to charge a non-scavenging power cylinder in accordance with the standard diagram Fig. 10, lb. per sq. in.

As determined from this equation, the maximum base pressure required for usual port characteristics is shown in Fig. 11, and it will be seen that it rises rapidly when the angle β_i is made less than 45 deg., especially so in the case of small stroke-to-bore ratios.

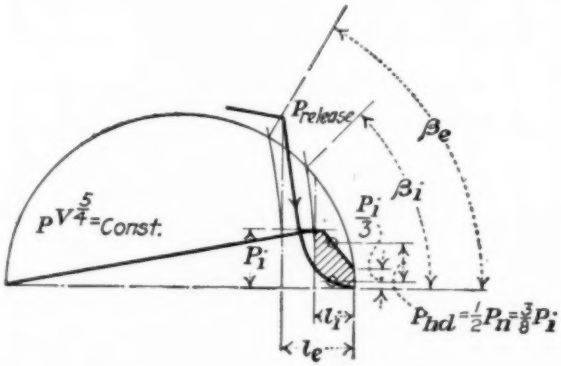


FIG. 10 STANDARD CHARGING DIAGRAM

40 The standard diagram requires the minimum expenditure of lost pump work after making due allowance for reserve speed contingencies, and on the whole it provides a most efficient basis for charging a non-scavenging two-stroke-cycle oil engine.

41 No advantage is to be gained by increasing the base pressure beyond the stipulated value given by Equation [11]. On the other hand, should the base pressure fail to reach the prescribed value, the rate of base discharge will be proportionately slower, which is likely to rob the power cylinder of combustion air and result in a considerable loss in rated engine power.

PUMP-WORK LOSSES

42 In Fig. 11 the limit line OA is empirically fixed at P_i = P.S./100, and this may be used as a practical guide for the

maximum base pressure allowable in a commercial oil engine of the non-scavenging type. When the base pressure greatly exceeds this arbitrary limit, the pump work lost in charging the power cylinder is likely to become excessive; as a rule the nozzle velocity V through the average intake-port opening should not exceed 600 to 700 ft. per sec. as fixed by Equation [**6**].

43 Knowing the maximum base pressure P_i, the accompanying pump-work loss may be estimated on the following basis: The pres-

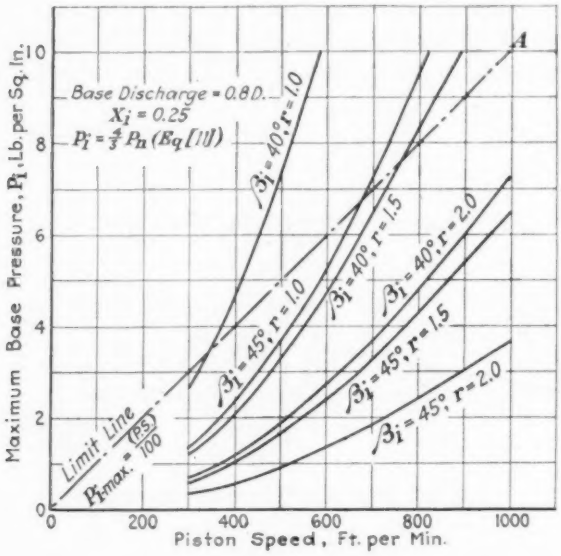


FIG. 11 BASE PRESSURE VS. PISTON SPEED

sure drop occurring during the suction stroke may be taken at from $\frac{1}{4}$ to $\frac{3}{4}$ lb., depending upon valve proportions. The compression curve for the base diagram may be found from the relation $PV^{\frac{5}{4}}$ = a constant, while the corresponding mean effective pressure for that portion of the diagram lying above the atmospheric line for a base pressure not exceeding 10 lb. per sq. in. is approximately $\frac{1}{2} P_i$.

44 The pump work lost in any two-stroke-cycle engine is most conveniently measured in terms of the full-load i.hp. produced within the power cylinder. Assuming the full-load m.e.p. to be 50 lb. per sq. in. for a well-designed non-scavenging two-stroke-

cycle engine of the semi-Diesel type, the corresponding values for the ratio of pump work to rated i.hp. have been plotted in Fig. 12. The limit line OA fixes the allowable pump work in an engine of this kind at about 10 per cent of the rated full-load i.hp. In order to hold to the given limit, the piston speed should not exceed about 750 ft. when $\beta_i = 45$ deg. and $r = 1.0$, or $\beta_i = 40$ deg. and $r = 1.5$.

45 The mechanical or friction losses in a two-stroke-cycle engine may be taken at from 10 to 15 per cent of the rated i.hp. developed

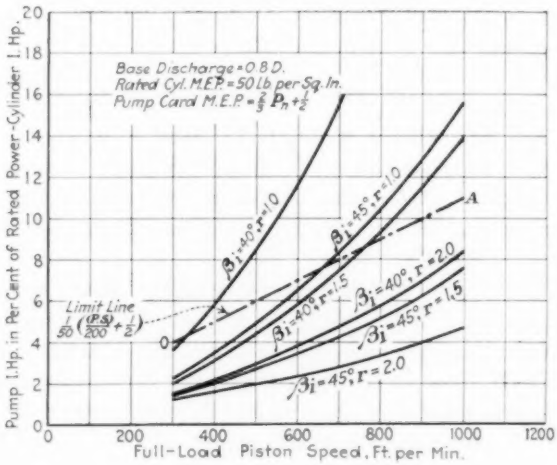


FIG. 12 PUMP I.HP. VS. RATED I.HP.

in the power cylinder. This loss, plus the specified pump-work loss, when deducted from the total i.hp. gives a fair measure of the expected rated b.hp. of the engine.

SCAVENGING CHARGING DIAGRAM

46 The preceding equations apply to non-scavenging engines for which the actual discharge from the base chamber is taken at $0.8D$. As compared to the four-stroke-cycle engine, the non-scavenging two-stroke-cycle engine is deficient in air content per unit of piston displacement, but this discrepancy may be largely overcome by the skilful use of scavenging air. However, when flushing the cylinder with scavenging air, care must be exercised in its use in order that the gain in indicated power will more than

compensate for the augmented charging loss. The scavenging-pump allowance is commonly fixed at about 50 per cent of the power-piston displacement.

47 In a scavenging engine it is advisable to prolong the pressure head behind the piston-controlled inlet port to a greater extent than is customary in the standard diagram of a non-scavenging engine. Fig. 13 shows a charging diagram suitable for a scavenging two-stroke-cycle oil engine. When running at full speed, the base pressure is preferably kept up to about $\frac{1}{2}P_i$ when reaching the dead-center position, which in turn increases the average effective pressure head maintained behind the intake port during the first β_i charging period to about $P_{hd} = \frac{2}{3}P_n$.

PISTON-DEFLECTOR-LUG PROPORTIONS

48 Not only should a relatively large volume of fresh air be charged into a two-stroke-cycle power cylinder during each and every revolution, but the greatest possible amount of this air should be trapped and held behind the piston at the instant of exhaust closure.

49 This result can best be achieved by effectively spreading the incoming air charge over a large portion of the cylinder area. In the case of piston-controlled intake ports, a large spread effect is most readily attained by diverting the flow of the incoming charge into the upper portion of the cylinder bore by means of a properly formed baffle plate cast upon the piston crown.

50 The intense eddy currents which are thereby set up, both at the baffle plate and the intake-port edges, can be made to absorb the major portion of flow energy initially given to the incoming charge. These eddy currents are especially effective in dissipating and slowing up the nozzle velocity immediately after passing the intake port and thus impart the desired wide flare to the long-sweep stream line on its way through the power cylinder.

51 The baffle-lug height h_i, shown in Fig. 1, may be empirically proportioned in terms of the intake-port length l_i. In best practice the ratio h_i to l_i lies between $1\frac{1}{4}$ and $1\frac{3}{4}$; an average value of $\frac{3}{2}$ serves to throw the charge well up into the cylinder bore and usually leads to good results in semi-Diesel engines.

52 When the height h_i is made too great, the incoming charge tends to flow too closely along the cylinder bore for best spreading effects. On the other hand, should h_i be made too small, the charge will initially spread out in fantail fashion, but upon further opening

of the intake port the charge tends to disregard the baffle lug and heads toward the open exhaust port. The radial depth d_i is also an important factor in securing the desired baffle effects and should be made equal to about $\frac{3}{4}l_i$ to $1.0l_i$.

53 The final proportions of the piston baffle lug for any particular type of engine are best determined by trial. An important aid in arriving at such proportions is afforded by a glass-top smoke model made to scale from a proposed cylinder design. A model having a 3 to $3\frac{1}{2}$ in. bore gives good results, and with it the probable path of flow may be traced for different lug proportions. The spread

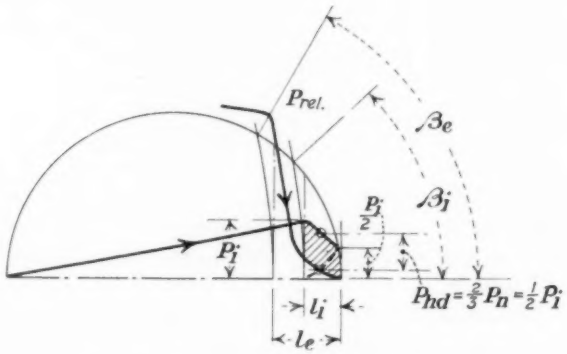


FIG. 13 SCAVENGING CHARGING DIAGRAM

of the incoming charge as it appears under the glass plate is illustrated by cross-hatched lines in Fig. 1. This spread area when multiplied by its mean depth serves as a fairly accurate measure of the air charge that may be trapped within the effective cylinder length S_e. The portion of the incoming air charge that is enclosed within the flaring stream line at the instant of exhaust closure, may be designated as the cylinder "filling."

EMPIRICAL FORMULAS FOR FILL EFFICIENCY

54 The "fill efficiency" of a two-stroke-cycle engine may be defined as the percentage of air content actually entrapped within the power cylinder per unit of piston displacement. A high power rating requires that this factor be kept at its maximum possible value.

55 A method developed by the author for estimating the air content that is likely to be trapped within the cylinder as a working charge is based upon a quantitive determination of the charge spreading effects possessed by eddy currents as measured with respect to the average intake port area. The reduction in nozzle velocity that can be effected by such means is found to be a logarithmic function of (*a*) the length of the flow path through the cylinder and (*b*) the ratio of the peripheral length to area of the intake port. By means of this method a series of interpolated values for fill efficiency have been found and the results plotted in Fig. 14. These curves apply particularly to two-stroke-cycle vaporizer engines working with an effective base discharge of $0.8D$. Accordingly, the following empirical formula may be used for roughly estimating the fill efficiency of moderate compression non-scavenging oil engines:

$$\frac{V_f}{D} = 1.9\sqrt{r}X^{\frac{1}{4}}\left(1 - \frac{B}{90}\right) = \frac{0.8}{K_n} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [12]$$

where V_f = effective cylinder "filling" or volume of preheated air charge confined within a piston-controlled two-stroke power cylinder at the instant of exhaust closure, cu. in.

V_f/D = fill efficiency, i.e., volume of preheated fresh air trapped within the cylinder per unit of piston displacement.

K_n = spill factor for a non-scavenging engine, i.e., ratio of effective base discharge of $0.8D$ to the volume of preheated free air entrapped within the cylinder.

56 Since both the intake and the exhaust ports of a two-stroke-cycle engine are open at the same time, not all of the air charged from the base chamber is likely to be held within the power cylinder. That portion of the charge which escapes through the exhaust port will be designated as "spill." As shown by Equation [**12**], the spill factor K_n varies inversely with the fill efficiency, and by the aid of this formula the spill applying to any given set of port characteristics may be approximated. It will be seen that a long stroke-to-bore ratio serves to improve the fill efficiency, but that this tends to drop somewhat with increased bore dimensions. The curves in Fig. 14 also show that an average fill efficiency of about $\frac{5}{8}$ to $\frac{3}{4}$ may be expected in a well-proportioned non-scavenging two-stroke-cycle oil engine of average size. The corresponding spill factor K_n therefore lies between 1.0 and 1.25.

57 The volume available for receiving the filling V_f may be termed "charge capacity" of the cylinder. In the case of a semi-Diesel engine working with a contracted vaporizer neck, its charge capacity is largely determined by the bore volume lying between the closing edge of the exhaust port and the mouth of the vaporizer neck, as designated by S_e in Fig. 1. The later in the stroke that the exhaust closes the smaller will be the charge capacity left to hold the incoming air charge, hence it is not advisable to extend the angle β_e beyond 60 deg.

58 Any surplus air in addition to the effective base discharge

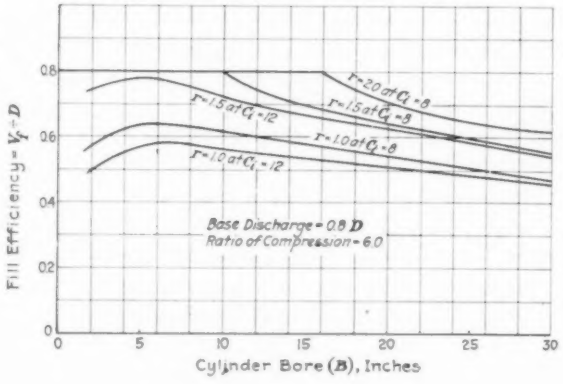


FIG. 14 FILL EFFICIENCIES

of $0.8D$ that may be blown through the cylinder offers better opportunity for the continued spreading of the incoming charge, and this causes the products of combustion to be more fully displaced by a cleaner and cooler working charge. The cleansing effect produced by such scavenging air may be taken as a logarithmic function of the augmented spill factor as shown in Fig. 15.

EFFICIENCY LOSSES IN SOLID-INJECTION OIL ENGINES

59 The various losses in fuel economy encountered in two-stroke-cycle semi-Diesel engines will be traced in some detail with a view of pointing out their sources and of indicating the direction along which further improvement is to be sought.

60 For rapid and perfect combustion it is essential that the injected fuel oil be thoroughly atomized at the spray nozzle, since

this greatly facilitates its even distribution throughout the air charge. This end is most readily attained when the spray nozzle is served by a fuel pump capable of "pulverizing" the liquid fuel and at the same time giving this finely atomized oil sufficient penetration to project it well into the surrounding body of compressed air. For best results, all of the air charge should be kept in close proximity with the spray nozzle.

61 As will be pointed out presently, most engines of the solid-injection type do not utilize all of the trapped air held within the power cylinder, since they fail to bring the entire body of air to

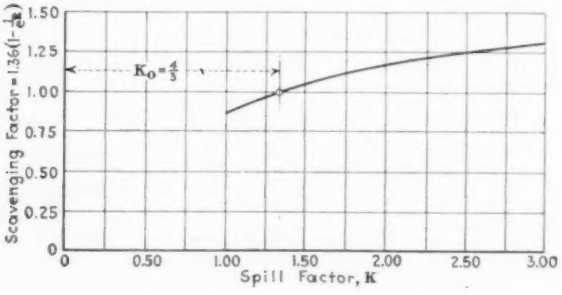


FIG. 15 SCAVENGING FACTOR VS. SPILL FACTOR

the support of combustion of the injected fuel oil. In case the oil is not thoroughly broken up at the nozzle but is injected in a jet-like spray, it becomes practically impossible to distribute the oil unless proper provision is made to evaporate the oil into vapor form prior to the ignition period. Furthermore, a relatively large portion of the air may lie dormant and lack the necessary turbulence to effect intimate contact with the atomized fuel oil.

62 When the injected oil is not evenly distributed and thoroughly mixed with all of the available air, pocketed or localized combustion takes place. In such cases, the feeding of additional fuel oil leads to imperfect combustion, notwithstanding that there may be more than sufficient air lying in isolated portions of the clearance space completely to burn such surplus oil.

63 For efficient combustion in solid-injection oil engines it is therefore essential to assist the mixing process by means of turbulence [1] within the working charge, the turbulence being created

[1] See Turbulence, by H. L. Horning, *Jour. Soc. Automotive Engrs.*, June, 1921.

either by high velocity of the incoming air or by means of piston compression. The disposition of the vaporizer and the extent to which the contracted neck of the vaporizer serves to create turbulence just prior to fuel injection have an important bearing upon the percentage of available air that can be counted on for the combustion of the fuel oil.

64 In a well-developed engine of the solid-injection type, fully $\frac{7}{8}$ of the filling air content entrapped within the power cylinder should be utilized for burning fuel oil. For engines lacking means for creating effective turbulence within the vaporizer, or for engines in which the spray nozzle is incapable of serving all the entrapped air, the available air for combustion of the fuel oil may easily be reduced to $\frac{3}{4}$ or less of the total air entrapped within the cylinder.

65 To obtain smokeless combustion, an oil engine requires a considerably larger excess of air than does a gas or gasoline engine. Taking the air required for a correct full-load explosive mixture in an oil engine at about 50 per cent in excess of the air theoretically necessary for complete combustion at best efficiency (as against 25 per cent in the case of a gas engine), the ratio of air to oil by weight becomes $15 \times \frac{3}{2} = 22 : 1$. Making a fair allowance of $\frac{1}{8}$ for deficiency in turbulence in the case of a well-designed solid-injection engine of either the two- or four-stroke-cycle type, the oil-burning capacity of such engines at full rated load becomes equal to $\frac{1}{22} \times \frac{7}{8} = \frac{1}{25}$ of the air weight confined within the vaporizer or that may be otherwise brought within effective range of the spray nozzle. Owing to the mechanical clearance required between the piston and the inner cylinder head wall, only about $\frac{3}{4}$ to $\frac{7}{8}$ of the entrapped stream-line air can be forced into the vaporizer chamber, and it is rather difficult to make all of such external air contribute its proper share toward the support of combustion of the injected fuel oil.

66 In the case of a full Diesel engine, its undivided air charge and the greater amount of turbulence obtained by means of the high-pressure injection air allow about $\frac{1}{22}$ of the stream-line air weight to be counted upon as forming its full-load oil-burning capacity. In addition, allowance should be made for the oil-burning capacity of the surplus air used to inject the fuel oil.

67 On the basis of 50 per cent excess air stipulated for efficient combustion, the ultimate, or overload, oil-burning capacity of a solid-injection oil engine may be taken at about $\frac{4}{5}$ of its full-load oil consumption. When the excess air supply is thus consumed, the exhaust is likely to become smoky, and, due to the resulting loss

in efficiency, the indicated power at overload will probably not exceed $\frac{5}{4}$ of its rated full-load power.

68 Taking the average value for fill efficiency of a two-stroke-cycle engine at $\frac{2}{3}$ and the density of the preheated air confined within the stream line at about 0.065 lb. per cu. ft., the full-load oil-burning capacity for a medium-sized non-scavenging semi-Diesel engine works out as follows: Weight of air trapped within the cylinder per cu. ft. of displacement is about $0.065 \times \frac{2}{3} = 0.0435$ lb., which in a solid-injection oil engine should be capable of burning about $0.0435 \div 25 = 0.00175$ lb. of fuel oil at its full rated load. When the lower heating value of the oil is taken at 18,500 B.t.u. per lb., the full-load heat consumption should be approximately $0.00175 \times 18{,}500 = H =$ 32 B.t.u. per cu. ft. of displacement, assuming all of the trapped air to be brought to the support of combustion.

69 The corresponding full-load indicated thermal efficiency is dependent upon the ratio of expansion and may be determined thus:

$$E_i = 1 - \frac{1}{R^{m-1}} \quad \ldots\ldots\ldots\ldots [13]$$

where E_i = indicated thermal efficiency at full rated load when working with properly atomized fuel oil at about 50 per cent excess air

R = ratio of compression as measured with respect to the exhaust port closure.

m = exponent for the expansion line PV^m = a constant. For oil engines of the vaporizer type, m should equal about $\frac{6}{5}$.

This value of E_i represents the highest indicated efficiency to be expected in a commercial oil engine for any specified compression ratio R. In case the oil is imperfectly atomized, or if the nozzle should work with excessive dribble, the given fuel consumption may increase by 10 to 20 per cent, or more. A non-scavenging two-stroke-cycle oil engine may be expected to reach at least $\frac{7}{8}E_i$ as fixed by Equation [13].

70 The full-load heat consumption H is a simple function of $\frac{\text{m.e.p.}}{E_i}$. Taking $H = 32$ B.t.u., as found above for a non-scavenging engine, and assuming that substantially all the trapped air is brought into combustion, the expected full-load m.e.p. works out to about

$$\text{M.e.p.} = \frac{H}{0.185} E_i = 175E_i \quad \ldots\ldots\ldots [14]$$

In this case the value of E_i is to be derived from Equation [**13**], and it will be seen that in a high-compression engine the attainable rated m.e.p. is increased because of its greater indicated efficiency.

71 At a value of $E_i = 28.5$ per cent, corresponding to a compression of about 130 lb., the m.e.p. at rated load for a solid-injection non-scavenging two-stroke-cycle oil engine at best works out to about 50 lb. per sq. in. This value is somewhat higher than is commonly reached in practice because not all of the stream-line air is brought into the combustion zone. In a well-developed engine of this type such loss of air should not exceed $12\frac{1}{2}$ per cent, thus reducing the corresponding rated m.e.p. to about $\frac{7}{8} \times 50$ or about 45 lb. per sq. in., i.e., its (m.e.p./E_i) ratio should readily reach a value of 160 when operating at best efficiency. A ratio as low as 125 to 150 is not uncommon for many engines of this class. While some two-stroke-cycle oil engines fail to deliver the specified m.e.p., the cause for this failure is to be looked for in faulty design of the vaporizer, defective atomization of the fuel oil, insufficient base pressure, improper port relations, and especially the inability to prevent localized combustion due to a lack of proper turbulence. Under favorable conditions the (m.e.p./E_i) ratio in the case of two-stroke-cycle scavenging engines may be brought up to 190 and better, as compared to 190 to 210 in the case of solid-injection four-stroke-cycle engines working with a single spray nozzle opening.

HEAT-FLOW LIMITATIONS

72 In addition to the speed limit imposed by excessive base pressure, a two-stroke-cycle engine should be checked for its speed limit as fixed by the allowable heat flow through the cylinder walls.[1] The maximum rotative speed at which an oil engine may be safely run is largely determined by the temperature assumed by the bore wall. The allowable piston speed for a line of two-stroke-cycle vaporizer oil engines should increase more or less regularly with the stroke dimension and in stationary, marine and other moderate-speed engines this is usually limited to (P.S.) = 175 to $200\sqrt{S}$. A high rate of heat flow is especially destructive in larger engines, and in order to hold the heat flow in a non-scavenging two-stroke-cycle oil engine below the limit set for safe and reliable running, its piston speed should not exceed 750 to 850 ft. per min.

[1] See author's Heat Flow Through Cylinder Walls, Trans. Soc. Automotive Engrs., 1918, p. 162.

CONCLUSION

73 The further development of solid-injection or semi-Diesel two-stroke-cycle oil engines should be looked for along the following lines:

a More perfect atomization of fuel oils, and prevention of dribble at the spray nozzle. The atomization of the heavier oils is much improved by the maintenance of high pressure behind a suitable spray nozzle. The fuel pump is the "heart" of a solid-injection engine and for perfect atomization, it should be capable of sustaining a high nozzle velocity throughout the entire spraying period at all running speeds of the engine. The drip formation may be effectively eliminated by abruptly breaking the nozzle pressure at the end of the spraying period.

b The vaporizer chamber should be kept reasonably small and should be designed to give the maximum possible turbulence to all parts of the working charge at the time of fuel injection, so as to compensate for the absence of high-pressure air injection. It is essential that the maximum possible portion of the air content entrapped within the cylinder be brought to the support of combustion of the liquid fuel.

c More careful design of the nozzle and baffle-lug elements of the piston-controlled intake port. By properly deflecting the flow path, and working for a larger spread of the incoming charge, it should be possible to improve the fill efficiency of a two-stroke-cycle oil engine and ultimately carry it beyond that now regarded as an adequate performance.

d Another promising, but as yet not fully exploited, mode of increasing the power capacity of a two-stroke-cycle cylinder lies in the more extended use of scavenging air. The relatively small loss of power involved in augmenting the air charge by means of scavenging pumps is more than compensated for by the gain in effective shaft power.

74 When the additional elements required to pump the scavenging air are kept reasonably simple and accessible, their cost may readily be absorbed by the resulting increased power rating of the engine without advancing its first cost per unit of output.

The greater cleanliness of the scavenged working charge results in more perfect combustion and improved thermal efficiency.

75 The author desires to express appreciation to Mr. D. O. Barrett for port data supplementing his article on Port Design for Two-Cycle Oil-Engines as published in *The Gas Engine*, February, 1919. He is also indebted to Mr. Jerome S. Marcus for his criticism of the manuscript and for a number of suggested changes that have been embodied in the paper.

o. 1807

COMPOUNDING THE COMBUSTION ENGINE

By Elmer A. Sperry, New York, N. Y.
Member of the Society

To engineers versed in the problems of selecting and designing prime movers, the advantages of the compound combustion engine are readily apparent. It is light compared with the normal Diesel, being in special cases less than one-tenth, and in some instances less than one-twentieth, the weight for the same output. Its mechanical efficiency is extremely high, and a distinct gain in overall efficiency from fuel to shaft has been made, as well as a very definite gain in simplicity, direct performance and smoothness of the crankshaft diagram. This has been achieved while adhering to the best practice, namely, four-cycle operation.

This paper presents the results of research by the author extending over a series of years, during which not only has the high-pressure principle been thoroughly established, but all the important requirements have been worked out, and finally an engine embodying practically all the advantages has been subjected to long continuous runs.

It is the author's belief that the results here announced for the first time warrant the prediction that engines employing the multi-stage or compound principle will at no distant date occupy a dignified place in the combustion-engine art. If this principle should assume leadership, it will of course only be because of its demonstrated practical usefulness and general merit. At present these appear to be of such magnitude as to at least insure wide adoption.

THE high-compression or Diesel cycle in combustion engines has worked nothing short of a revolution, having brought to the prime mover its choicest heritage, the highest thermodynamic efficiency known. The fuel economies of these engines have forced them to the front. They have become extremely reliable and easy to operate; instances are becoming common of long runs without overhaul — long-continued performance without shutdown or forced stop of any kind. There is one record with a marine Diesel of 27,000 miles without trouble or adjustment. Another engine ran 18 months, 24 hours daily, with 2 hours' shutdown each Sunday morning for inspection, no part being replaced during this time. A Diesel ship

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of 8000 tons, 2500 i.hp., has run continuously for ten years; her mileage is more than 500,000 and she is still "going strong"; her lay-ups for repairs to machinery have been small and on the average less than they would have been with steam equipment, there being no boilers to reckon with.

2 Beginning with this particular ship, there are now 325 Diesel-engined ships, representing a deadweight tonnage of more than 1,250,000 tons ocean-going. Again, about 90 per cent of the cancellations during the recent depression have been for steam vessels. At first steam engineers refused to operate Diesels in ships, but this is now all changed and motorships are in such high favor that they are leaving steam vessels and seeking jobs on Diesel ships. All this emphasizes the fact that dependable, high-efficiency, oil-burning combustion engines have arrived.

COST AND WEIGHT OF DIESELS HIGH

3 As experience is gained with these engines, however, there have developed some objectionable features which are serious. Incidentally the three-stage high-pressure air-injection pumps are more or less sensitive and require considerable experienced attention, but what is really serious is the great weight of the engines and the heavy capital charge or first cost. Though they occupy somewhat less space than boilers and engines, the weights of Diesels are on a par with, if not somewhat in excess of, those of reciprocating engines with their boilers and decidedly in excess of those of water-tube boilers and turbines. Diesels now in the merchant service illustrate this. The standard product of the largest builder of these engines for the merchant service weighs about 450 lb. per shaft horsepower and is large and bulky. A substantial increase in tonnage of freight carried is only one of the gains that would be secured, could these engines be made much lighter and smaller for the same power.

4 The weight is not the only difficulty with these large engines, however, for they cost more than steam equipment of equal power. One instance of $200,000 excess for a 3500-s.hp. ship is cited. In another ship for the same service and of the same power the weight was 60 per cent in excess of the turbine equipment and the cost 212 per cent, amounting to $306,000 excess. Notwithstanding this extra capital charge, the first ship at three-fourths capacity in the Far East trade can earn nearly double net (83 per cent) over its

turbine competitor of the same power and construction and with its machinery weighing a third more than steam.[1]

SPECIAL FUELS REQUIRED BY DIESELS

5 Again, it has been found that almost without exception a grade of fuel oil known as Diesel oil must be employed. This is a partially refined product costing upon the present market considerably more than bunker oil burned under the boilers.

6 The differential price in Diesel over bunker fuel is large and is a handicap destined to grow more serious as the demand for gasoline for automotive purposes requisitions more and more of this Diesel fuel for cracking. A disciple of very heavy Diesels has proposed to put bunker oil through a conditioning process to be fitted for use in Diesels, employing a percentage of the more refined and expensive fractions. This, again, would be a serious handicap. The successful engine must use a wide range of the cheap fuels picked up in any market, "as is," without alteration or adjustment of the engine.

7 The fuel-oil problem in itself renders some advance absolutely imperative. Conservation of our oil should be backed by Government enforcement to stop the prodigal waste which results from bulk or furnace firing of enormous quantities of these highly concentrated fuels, destroying three to five times the quantity, power for power, required by steam, especially now since the cheapest petroleum can yield to the full its wonderful store of energy in the most direct way possible by being burned drop by drop directly in the cylinder and practically at the point where the work is to be done. This is the way, and the only way, our vital heritage in petroleum should ever be allowed to be consumed for power purposes, where it makes its fourfold and maximum contribution.

8 Now let us look into the possibility of overcoming the two other principal difficulties, viz., the capital charge and the excessive weight and size of the engine.

COMPOUNDING AS A REMEDY FOR INHERENT DIFFICULTIES WITH DIESELS

9 In a report made by a group of engineers in 1900, it was stated that if the combustion engine could be successfully compounded, a most important gain would be made in its weight and

[1] See paper by Metten and Shaw before Society of Naval Architects and Marine Engineers, May, 1921.

size. The fact that compounding presents other advantages has been known to engineers for a number of years, but the difficulties have been looked upon as insurmountable. The fact that Diesel himself in the '90's undertook the construction of a compound, the total failure of which received wide publicity, reports of these failures finally getting into textbooks, seemed to give substantial corroboration to this point of view. One eminent German authority, Güldner, set the final seal of "impossibility" upon it with characteristic Teutonic finality in the following statement:

> As it was, the entire motor proved to be a flat failure, as was and will be the case in any compound gas engine in spite of the best care and enormous expense.

10 This statement has probably done more to retard combustion-engine advance, especially in this important phase, than anything published in the last two decades. A close study of Diesel's embodiment, in view of present knowledge, reveals a total lack of most of the features that we now recognize as essential to success. There have been one or two isolated workers in England and Germany who have not been so ready to believe this statement, feeling that the next great step in combustion engines, as in the case of steam, is bound to be compounding, but practical difficulties still intervened and no advance was made or even serious work done until within the past couple of years. Attempts in England have met with the most discouraging and unwarranted opposition. Authorities of experience who should be open-minded, while recognizing the advantages in part, condemn compounding apparently through sheer refusal to depart from beaten paths and emphasized certainty that the difficulties of temperature and losses at transfer can never be overcome.[1]

11 A year ago Professor Watkinson, director of the important engineering laboratories of Liverpool University, in discussing an epoch-making paper by Engineer-Commander Hawkes on the Admiralty's extensive research on Diesels, stated, in substance, that we must recognize that the combustion engine in its present state was very crude and in compounding only lies the line along which the next great step in progress would be made. And he also stated that much greater results are bound to follow the compounding of combustion engines than was ever realized by compounding steam

[1] See discussion of Chorlton's paper on Super-Compression and Economy of Internal-Combustion Engines, *Engineering* (London), July 8, 1921.

engines; first, because of the very much greater range of pressures available, and secondly, because the great enemy to compounding in steam, viz., cylinder condensation, is not present.

THE AUTHOR'S EARLY WORK IN COMPOUNDING

12 Activity in this field is prodigious and workers in various parts of the world are commencing to realize that a great advance step is imminent. In America, a group headed by the author has been engaged on this problem for upward of thirty years. Starting in 1890, his first compound was running before the World's Fair. The patent records give evidence of this early work under date of December 10, 1892. A number of engines have steadily followed each other, each involving improvements resulting from previous experience, until the essential problems have been conquered. Not only has the principle been thoroughly established, but all the important refinements have been worked out, and finally an engine embodying practically all the advances has been subjected to long continuous runs. Data believed to be almost invaluable to the art have been secured, together with a series of indicator cards and diagrams that exceed a thousand in number.

13 The outcome has been that the various prophecies of thoughtful engineers in the past have been more than fulfilled and there is every evidence that the heavy-duty compound combustion engine is everything that was hoped for. It is light compared with the normal Diesel, being in special cases less than one-tenth, and in some instances less than one-twentieth, in weight for the same output. Its mechanical efficiency is extremely high, a distinct gain in overall efficiency from fuel to shaft has been made, as well as a very definite gain in simplicity, direct performance and smoothness of the crankshaft diagram. This has been achieved while adhering to the best practice, namely, four-cycle operation.

COMPOUNDS LIGHTER AND CHEAPER — MUCH LARGER UNITS POSSIBLE

14 The lightness and simplicity of the compound solves the capital-charge factor automatically. Engines and machinery in the final analysis usually cost a given amount per pound. It is found that there is no structural demand or refinement in the compound that has not already been met in the Diesel — the same range of pressures and temperatures are handled throughout by the effective two-stage method, which is now well established. Engines of this

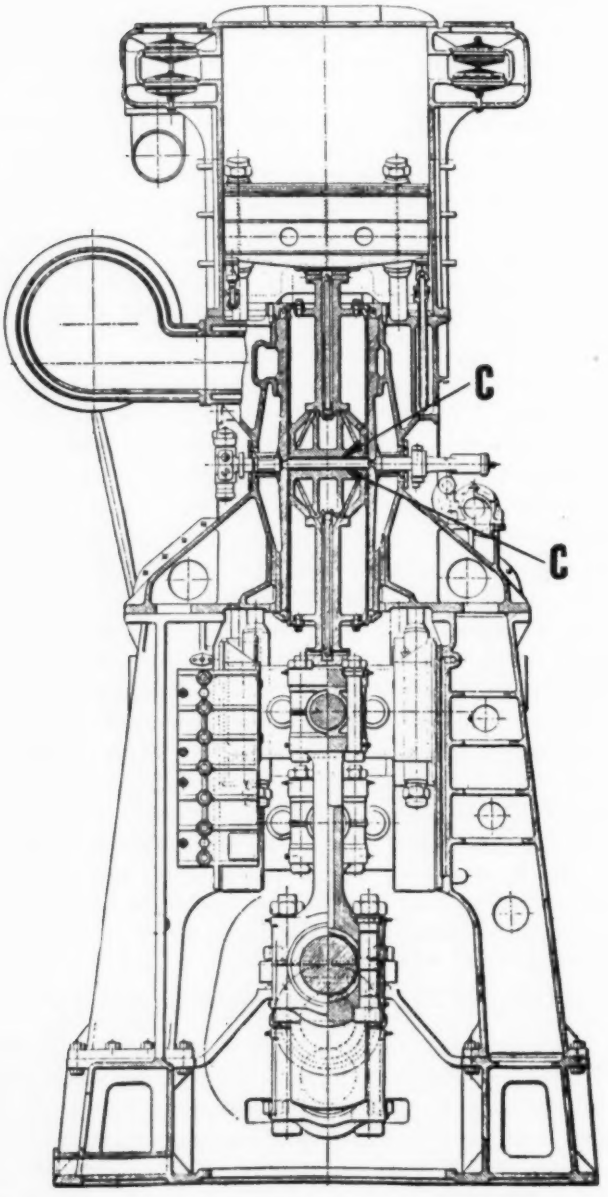


FIG. 1 SECTION OF JUNKERS ENGINE SHOWING COMBUSTION SPACE

type weighing only a fraction of the weight of the present Diesel will inevitably be found to be much less in first cost as well as in cost of upkeep.

15 With light, satisfactory combustion engines for ships, for instance, all question of their being better adapted to long hauls than to short hauls, to this service or to that, disappears if their first cost is sufficiently low. They are then best for all kinds of service and become necessary to the economical operation of ships generally.

16 Much higher powers than are now available are thought to be of extreme importance and much interest is centered upon the question; in fact, the more advanced among the Diesel builders are today concentrating upon ways and means to attain still higher powers from a single cylinder working on its present cycle. Of course this will increase instead of decrease the weight per horsepower. It is to the compound that we must look for a solution of this problem, for the reason that within present cylinder and cylinder-wall limitations, powers in excess of 10,000 hp. per engine are entirely practicable and are accompanied by all of the proportionate savings in weight, space and capital charge that have been pointed out.

17 Our own Government has watched the progess of this work for a number of years, inspected the development in its various stages, and has come forward with orders for an initial engine which, together with orders from other sources, is now under construction. To illustrate to what low figures the compound principle can be relied upon to bring the heavy-oil engine, it should be stated that among these orders is one now under construction to weigh about five pounds to the brake horsepower. This may be looked upon as extreme, but the designed weights and finished parts as they now stand are below this figure. This brings us within striking distance of aviation engines, where the fire risk through the presence of gasoline and the electric ignition system constitutes one of the greatest menaces to aviation progress. The remarkably high mean effective pressures of the heavy-oil compound will give us the aviation engine and entirely eliminate both these sources of fire risk.

REASON FOR GREAT WEIGHT IN DIESELS

18 The reason for the great weight in all classes of Diesel engines is apparent when it is understood that they are designed around an extremely small quantity of air and oxygen at each stroke,

an amount so minute as to be quite surprising when compared to the ponderous engine itself. The volume of the power gases available to do work is limited to the size of the combustion space, and when it is remembered that this is confined to a small crevice in the end of the cylinder of the order of one-twentieth of the travel of the piston, the limitation is at once seen. Attempts to make it larger produce both compression and ignition temperatures that are too low and we have semi-Diesel and surface-ignition engines confined to smaller sizes with tendencies to a lower grade of performance. Such engines are difficult to start and represent no gain in weight.

19 Two of these volumes are sometimes combined as in the opposed Junkers engine. The size of the combustion space, though still minute, as shown in Fig. 1 (notice clearance volumes *CC*) is known to give some small advantages over the regular engine which has only one-half of this space, as at *C*, but has the disadvantage in common with all two-cycle engines of the fractional piston-ring

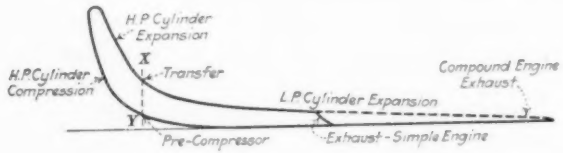


FIG. 2 INDICATOR CARD OF DIESEL ENGINE

support practically without lubrication on the hot bridges as the exhaust port is uncovered by the piston. These latter require an added structure in the form of a large scavenging and air-supply pump, connections, valves, etc. They are also very heavy. An apparent four-cylinder opposed engine is really an eight-cylinder engine requiring twelve cranks. (See 1914 German Junkers and 1921 Doxford.)

SEPARATE CYLINDERS SUITED TO PRESSURE RANGES — TWO-STAGE COMPRESSION SECURES LARGE WORKING VOLUMES

20 Take, for instance, the Diesel indicator card, Fig. 2. Draw the vertical line *XY* and it is easy to see that the expansion pressures to the left are high and should do their work in a high-pressure cylinder, whereas those to the right, especially following the dotted extension line of expansion, are definitely low pressures and should do their work in a low-pressure cylinder suitable for the purpose.

This is practically all that is attempted in compounding combustion engines. The vertical line XY also divides the compression curve into two stages, low and high, the latter taking place in the combustion cylinder proper as a second stage. No compressor delivering 500 or more pounds per square inch will undertake to do so in a single stage; there would be at least two stages. The old single-stage compression is discarded in the compound and this modern method of two-stage compression is adopted.

CHILLED SURFACES REDUCED

21 Supercharging or compressing in two stages gives the controlling advantage in that a very much larger unit volume of gases may be handled. The clearance spaces may be many times the size of those in the Diesel, and yet it is perfectly simple to bring these large volumes up to the requisite pressure and incandescent temperatures at the instant of fuel injection.

22 The large volume of power gases in the combustion chamber of the compound at once solves a number of important problems, makes the light engine easy of accomplishment, and overcomes a number of other difficulties at the same time. No longer is the chilled perimeter per unit volume of gas the controlling factor, as it is in the Diesel. The chilled walls are retired into the background as the large volume asserts itself, as compared with the small crevice indicating the total volume available in the normal Diesel (shown at C in Figs. 1 and 14), where the chilled areas exposed are very large and the volume very small. At a glance it will be seen that all this is reversed completely in the large dome-shaped clearance space D in Fig. 14, where the volume has increased very much more rapidly than the perimeter. This is again vastly increased in the low-pressure cylinder, where an extremely large volume exists with still smaller ratio of chilled perimeter. Taking all of the chambers into consideration, it is found that while retaining all of the chilled walls that are necessary for proper handling of the lubrication, still a gain is made on the order of 60 per cent in the value of these chilled walls in the compound as compared with the simple engine.

SOLID FUEL INJECTION

23 With the large clearance volume we no longer have difficulty with solid injection, nor do we have any difficulty in using a wide range of heavy fuels. It has been known for years that as soon

as the oil spray encounters chilled or even red-hot walls, the efficiency drops.[1] Here the oil fog may penetrate the deep masses of hot compressed air with instantaneous effect in every direction from the spray nozzle without encountering chilled walls and instantaneous and very complete combustion results. In the compound engine the clearance volume is so large that the entire high-pressure piston displacement causes it to lose only a fraction of its pressure, thus bringing to the second stage, or low pressure, both ample volume and pressure so that this piston (representing 6, 8 or even 10 times the area of the high-pressure) is driven to the end of its stroke with pressures still above the atmosphere.

MUCH GREATER EXPANSION, HIGHER EFFICIENCIES

24 In this way the engine yields an expansion ratio based on gage pressures, which instead of being 3 or 4 to 1, as in the case of the automobile engine, or about 12 to 1 in the Diesel, can be made as high as 120 to 1. Yielding a higher return and greater efficiency from the fuel because of the lower temperature of the exhaust, similarly the high-vacuum steam turbine secures higher efficiency by its much greater ratio of expansion as compared with the reciprocating engine. The great volume in the combustion space furthermore allows this space, without distortion, to extend easily out over the top of the low-pressure piston, making a most direct connection therewith through a short transfer port (see *L*, Fig. 14). And finally this large clearance space allows its shape and contour to be grouped almost solely with reference to the most complete dissemination of the fuel and highest combustion efficiency.

25 In aeronautics, supercharging has given America the official "ceiling" of the world through greatly increasing the powers of the aero engine, but the process, *per se*, is wasteful in forcing higher exhaust pressures and temperatures. In compounding we have these wastes conserved by coupling supercharging with superexpansion, securing the maximum advantage of each.

THE COMPOUND CYCLE

26 Fig. 3 shows the cycle card without fuel, taken from the high-pressure cylinder in a 10:1 compound. The first-stage compression enters the cylinder at point *A* on its out stroke with the pres-

[1] See Eng-Comdr. C. J. Hawkes' Admiralty paper: Experiments . . . Diesel Engines, November, 1920.

sure about 113 lb., giving a power stroke indicated by line A'. At point B the induction valve closes and on the in stroke the compression proper starts and rises on line B' to Diesel values at C. There being no fuel injection, the receding stroke brings the pressures down on practically the same line B' to point B (E in Figs. 5 and 6). Here the transfer valve opens and the gases pass on the in stroke to the low-pressure cylinder on line D. This is the out

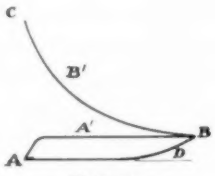


FIG.3


FIG 4

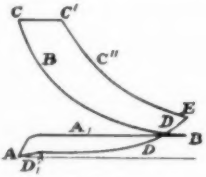


FIG.5

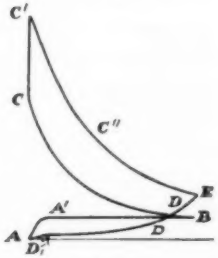


FIG.6

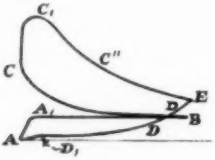


FIG.7

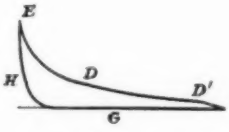


FIG.8.

FIGS. 3–8 INDICATOR CARDS OF COMPOUND CYCLE

stroke of the low-pressure, the pressures sinking to a trifle below atmosphere before the final exhaust valve opens and continues open nearly during the entire next stroke of the low-pressure piston, which is, however, never shown on this high-pressure card but on the low-pressure card, Fig. 8.

27 Fig. 4 is the air-pump card representing the first stage of the compression which delivers air to a small receiver, from which air is delivered during the induction stroke, on line A' of Fig. 3,

which, it will be observed, is a power stroke with a high mean effective pressure, thus recovering some of the power of the pump.

28 Fig. 5 is the same as Fig. 3, except that fuel has been injected, and shows the regular slow Diesel burning common to all early cards of these compounds, where from point C a perfectly level line is often drawn to point C' which marks the point of "cut-off." The gases then expand on line C'' to point E, where the transfer valve opens and the gases continue to do work on the large area of the low-pressure piston, indicated by line D. The exhaust valve opens with pressures only slightly above atmosphere at point D'. The ordinary slow Diesel burning has the objection of lower efficiencies and allowing the heat to be added to the gases out nearer the exhaust end of the stroke.

UTILIZING THE DETONATION OF FUELS

29 Through a research extending over a year and a half or more, conditions were discovered by means of which "detonation" of the fuels may invariably be secured. This extends to a large variety, including of course the heavy fuels. While automotive engineers have adopted extreme measures, even lowering compression, "doping" fuels, etc., etc., in order to avoid the detonation, we have been working in the opposite direction. During the war the Admiralty entered upon intensive research into many phases of Diesel performance with full-size single-cylinder engines of a number of types. Those familiar with these extensive operations cannot speak in too high terms of the ability and experience brought to bear on the many vital questions investigated and the very great importance to the art of the results that rapidly followed each other. These and coördinated activities are considered now to place England easily in advance of Germany and other countries in the matter of Diesel performance and construction. The author has already mentioned two papers by Eng-Comdr. Hawkes covering some features of this work, in the course of which detonation was encountered which he characterized as "rather distressing" under certain circumstances. He goes on to say: "It was very desirable, therefore, that every effort be made to eliminate the tendency to detonate," but he reports that it was found impossible to do this without at the same time increasing the fuel consumption. The author, however, found that the *thermodynamic efficiency was higher in case of high detonation diagrams* than with low. Our work has

been toward realizing these high efficiencies by developing instead of suppressing this high-intensity combustion. We have therefore sought to harness and utilize to the full the detonation phenomenon for the reasons and with the results outlined herein.

30 Fig. 6 is a typical card taken under the same conditions and the same engine as Figs. 3 and 5, exhibiting "detonation" and showing the quick burning, piling the heat up at the beginning of the stroke and as far away as possible from the exhaust end, giving also a very much truer Carnot-cycle card. All recent running of the compound engines has been in accordance with this card. One of the achievements of high-intensity combustion is better thermal efficiency and a still further reduction in the exhaust temperatures. The engine with this type of card is lifted from the old constant-pressure to the superior constant-volume performance which frees the compound from all speed limitations. The detonation characteristics, e.g., the vertical rise on the diagram, persist even at very high speeds. This card also insures operation at lower fractional powers without indication of carbon deposits.

31 Fig. 7 is given as showing the range of control of the peak and shape of the card at will under the same operating conditions and with the same spray nozzle.

32 Referring to Figs. 6 and 7, the peak pressures given by these cards may seem high, but it must be remembered that they are not as high as the pressures often met with in the cylinders of many gasoline engines. The factors of safety easily take care of these. Ample provision, however, is made as the engine is already so light that plenty of leeway is available for any of these purposes.

ELIMINATION OF LOSSES IN TRANSFER

33 In early attempts at compounding, principally layouts, that have come to light, it was found that prohibitive losses would occur in the transfer, due to the falling pressures while filling the low pressure clearances. A complete solution of this is found in a special adaptation of the process of "cushioning" — closing the exhaust valve at a predetermined point before the out-stroke end, trapping a little of the hot gases and cushioning them up to the transfer pressure so the transfer valve opens under conditions of equal pressure on each side. There is therefore no flow of gases except that due to the slow starting of the strokes of the two pistons. There are practically no losses sustained in cushioning;

the power of compression is returned very completely on expansion. In this unique adaptation of cushioning the additional advantage is secured of preventing all erosion due to high velocities of the hot gases over the transfer seats. These seats are amply jacketed and are found to remain smooth, bright and perfectly sealed over long periods. It is incidentally found that in cushioning, the adiabatic compression of the hot gases brings with it an equality of *tempera-*

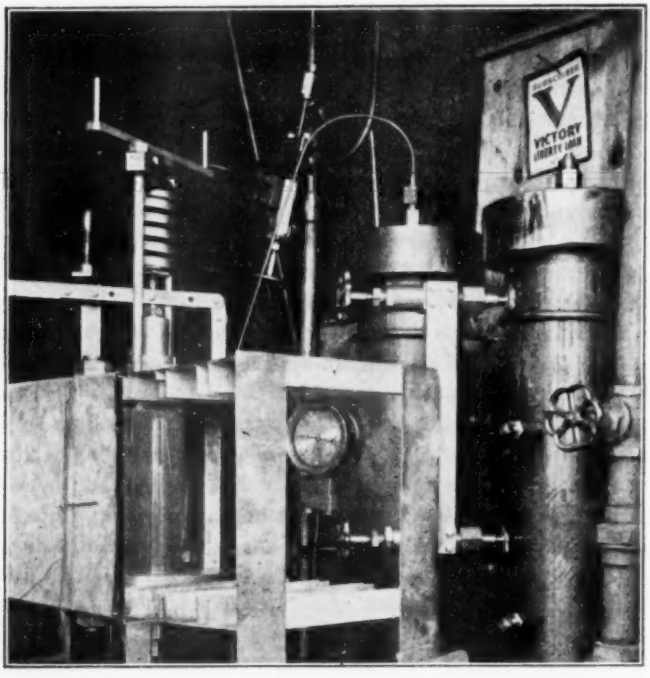


FIG. 9 RESEARCH APPARATUS DESIGNED TO REVEAL ACTUAL CONDITIONS WHEN OIL IS INJECTED INTO A GLASS CHAMBER UNDER PRESSURES ABOVE 600 LB. PER SQ. IN.

tures as well as pressures, so there is neither loss in pressure nor temperature at this critical point of transfer and the efficiencies are carried at high values throughout the cycle. In the compound engine the exponents hold very closely to the adiabatic throughout the cycle due to less cooling, owing to the very low ratio of chilled wall surfaces to unit volume of power gases and the large unit volumes employed.

34 Fig. 8 shows the low-pressure card, indicating the same

power-pressure line D as Figs. 3, 5 and 6. On this in stroke the exhaust valve opens at D' and closes again at point G on the out stroke, giving rise to the cushioning curve H, terminating at the same pressure and really the same temperature as at E in the high-pressure cylinder.

35 Fig. 11 shows one of the early cards of the slow-burning

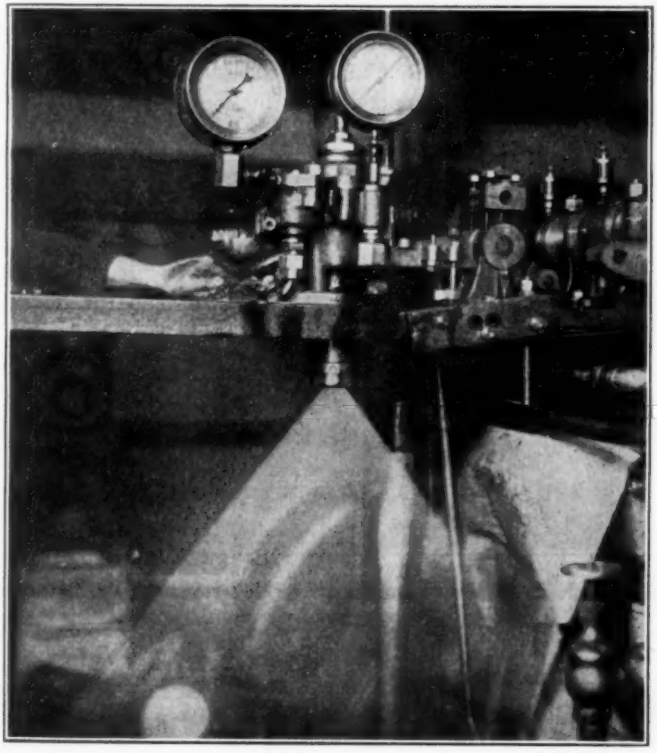


FIG. 10 ACTUAL SOLID-INJECTION JET WHICH PRODUCES DETONATION

Diesel type photographed here for the purpose of indicating the complete fuel control. The one to the left shows a great many expansion lines in both high- and low-pressure cylinders, obtained by varying the fuel in small gradations between the extreme limits of 10 per cent overload and idling. The card to the right is similar, the expansion lines being in two groups representing full load and fractional load.

36 Fig. 12 shows a number of superimposed detonation cards where the solid injection of the fuel is adjusted for about 10 deg. "lag," showing that the vertical rise constituting the detonation curve does not depend on "lead" for the instantaneous completion of combustion, as it does where detonation is not present. On this card the compression has risen to its peak and is clearly seen to have started down again quite perceptibly before the fuel is introduced and the vertical "expansion" line starts to rise. This line is vertical in a 7-in. by 11-in. engine running on the order of 1000 ft. per minute piston speed. The marked contrast between this vertical characteristic and the flat-topped cards in Figs. 5 and 11 is obvious.

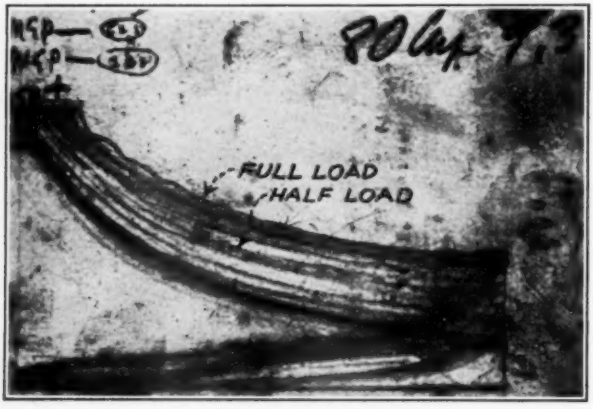


FIG. 11 AN EARLY CARD OF THE SLOW-BURNING DIESEL TYPE, INDICATING COMPLETE FUEL CONTROL

37 An important feature of compounding is the complete suppression of all preignition. Work by the Bureau of Standards [1] shows that no preignition rises to high or dangerous pressures in the combustion chamber which occurs late on the compression curve. Only when they occur in the early part of the compression curve are they combined with further compression and thus rise to dangerous values. Two-stage compression cures this completely because all the early part of the compression occurs entirely extraneous to the combustion chamber, where air only is handled, and only the latter part or second stage of the compression occurs in the com-

[1] S. W. Sparrow: Increase in Maximum Pressures Produced by Preignition in Internal-Combustion Engines, August, 1920.

bustion cylinder, so the engine is simplified because no troublesome and often leaky safety valves or pressure relief valves are required or fitted.

THE TRANSFER-VALVE PROBLEM

38 The point X in Fig. 2 will be recognized as being the same as point B in Fig. 3 and point E in Figs. 5 and 6. The transfer valve considered as an exhaust valve is here called upon to handle much hotter gases than ever heretofore. A difficult situation is here presented. Are special materials necessary and how can the valve successfully perform this duty? It is known that the exhaust valves

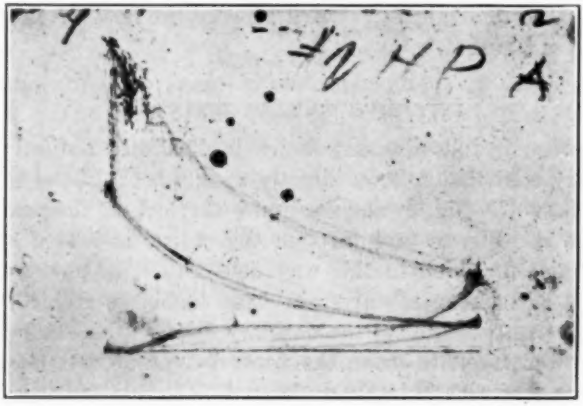


FIG. 12 SUPERIMPOSED DETONATION CARDS SHOWING THAT THE VERTICAL RISE CONSTITUTING DETONATION CURVE DOES NOT DEPEND ON LEAD FOR INSTANTANEOUS COMPLETION OF COMBUSTION

of the Liberty motor run red hot and the stems white hot, and yet they are handling gases of lower temperature than are here present. Nor is the Liberty motor by any means the only combustion engine where the exhaust valves operate regularly at red heat. These transfer valves may be water- or mercury-cooled. In fact, mercury-cooled valves were originally provided and have now stood on our shelves unused for some years as neither of these expedients is found necessary, nor are any but ordinary materials required. It must be remembered that compression in the compound is by the modern two-stage method. Air is admitted to the combustion chamber under comparatively high pressure and although it is warm, yet with each atmosphere of pressure its cooling powers are doubled. Air at

100 lb. thus has seven times the cooling power of atmospheric air, seven times the weight and seven times the molecules in contact for cooling. In forcing the high-pressure piston down on pressure line A' in the above figures, air must pass some port in entering. Now, as a matter of fact, this port is in line with the transfer port and the induction valve itself rides on the back of the transfer valve in the form of a hollow sleeve I (Fig. 14) seated directly on the top of the transfer valve T. The back of the transfer valve is provided with greatly enlarged radiating and cooling surfaces presented to this cooling air and powerful convection currents are constantly acting when sealed. Moreover, this air when entering is at high velocity and gushes down through and bathes the deeply serrated surfaces of the back of the transfer valve, licking up the heat very completely in its inward rush.

INTENSIVE COOLING EFFECTIVE

39 Now in following out the cycle, it will be noticed that this is the very step that follows directly on the heels of the transfer of the hot gases (D, Fig. 3) and continues throughout the next quarter cycle (see A', Fig. 3) and through the entire descent of the high-pressure piston, which in this way delivers a real power stroke to the crank with mean effectives in some instances greater than the mean effective pressures of the ordinary Diesel, thus returning some of the power taken to drive the supercharger or first stage pump. If the transfer valve is intensely heated on its under surface (see T, Fig. 14) and is then instantly intensely cooled on a surface five times as great, it will certainly strike and maintain a heat balance which in practice is found to be extremely low, only about one-half the temperature of the Liberty valves, nowhere nearly approaching red heat nor the temperature of normal Diesel exhaust valves under load conditions. Here we have a case, as Kipling says, "where the thing that couldn't has occurred" — it is the crux of successful compounding, the great obstacle which has always been looked upon as practically insurmountable, solved by an extremely simple and straightforward method. In fact, the temperatures of the valve are lower than the calculated temperature balance because of the high velocity of the induction gases from the receiver.

40 Again, the heat in these gases absorbed from the hot valve is useful inasmuch as it is the auto-ignition temperatures as well as pressures that are required at the end of the compression curve B'. Here a useful heat transfer and pure regenerative process is carried

out. The seats give no trouble because they are backed by the ample water jackets and, in fact, the whole transfer valve gear operates continuously and successfully and is found to be in perfect condition after hundreds and even thousands of hours of operation. They require practically no attention and are seldom ground in. This is is exactly as it should be, considering the very low temperatures at which this valve really operates.

HOW LIGHTNESS IS OBTAINED IN THE COMPOUND ENGINE

41 The question often asked is, to just what is due the smallness and lightness of the compound engine? It is this: In the four-cycle Diesel we have the tonnage of metal due to the presence of high pressures, operating at a ridiculously low material efficiency because these high pressures persist only about $2\frac{1}{2}$ per cent of the total time. The Diesel card rises abruptly and immediately falls. All the rest of the time, over 95 per cent, either low pressures or no pressures at all are present, whereas in the compound the pressures *persist* and we are dealing with great blocks of power, as can be seen from the power curve C''. Although the pressures are not materially higher than in Diesel practice, they are made to persist practically clear across the card, producing very large gross mean effectives. This is instantly followed by another line clear across the card, again producing another large gross mean effective in the low-pressure cylinder when referred to the high-pressure area, all from a single fuel injection. Instead of 60 to 70 lb. net mean effective to the crank, delivering its power through a few degrees only of one stroke in four, in the compound we have two net mean effectives, each of 300 or 400 lb. per sq. in., succeeding each other and covering two strokes out of the four from a single fuel injection, giving very much better crank-effort distribution for power purposes. The point of paramount interest is that these two large blocks of power are secured *not by any material increase of pressures*, but by using large quantities of power gases, and "hanging on" to the pressures we have in those gases throughout practically two complete strokes, clear across the card twice, thus abstracting much more of the power they contain before exhausting. Suppose these to be 330 lb. per sq. in. each. Added they make 660, which is easily ten times 62 lb., a net mean effective not infrequently met with in ordinary Diesels. In an engine of simple construction giving ten times the net mean effective to its crankshaft and well distributed, there should be no

good reason why it should weigh more than one-tenth the weight of many of the present Diesels.

42 The power gases work in the Diesel about 120 deg. of arc and in the compound 315 deg., or 2.6 times as long; or, considering the points of "cut-off" in each, the true expansion curve is $3\frac{1}{3}$ times as long, which accounts for its large mean effectives and higher economies.

PROPER RATIO FOR COMPOUNDS

43 As to the proper ratio for compounds; engines of 10 : 1 ratio of low-pressure to high-pressure cylinder areas, also 8 : 1 and 6 : 1 have been made, operated and studied, the smaller ratios being at present considered more desirable. The weight factor does not change materially with changes in ratio in this region. The

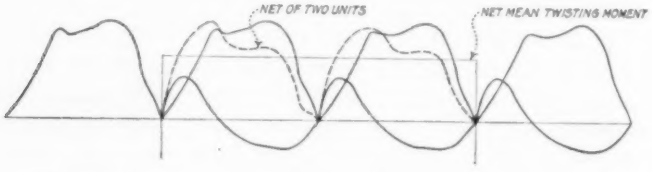


FIG. 13 COMBINED CRANK-EFFORT DIAGRAM FOR TWO COMPOUND DIESEL ENGINE UNITS WITH CRANKS AT 180 DEG.

low-pressure piston operates two-cycle. The power distribution and the weight of the reciprocating parts both equalize best at about 6 : 1. This makes a perfectly balanced unit, the end masses equaling and also moving oppositely to the central. The two full power impulses following each fuel injection are also about equal. Thus full *four-cylinder performance* is secured with only three cranks and two extra power impulses are delivered on the induction stroke (see line A', Fig. 3), making six power impulses for each cycle.

44 Another unusual advance should be noted, viz., complete reversibility and self air starting may be secured without additional valves or cams over the simple, one way engine without air starting, there being no difference in this regard. Again comparing the full-reversing, air-starting compound unit with a similar four-cylinder Diesel of any prominent make, delivering the same number of primary power impulses to the crank, the latter has 16 valves and 32 cams. The former operates the same cycle with two extra power impulses over the Diesel with 5 cams driving 7 valves.

THREE-STAGE COMPRESSOR AND BLAST REFRIGERATION ELIMINATED

45 Doing away with the three-stage air injection pump, its intercoolers and general complexity, is another important simplification. The use of high-pressure injection air introduced into the cylinder with the fuel is a step in the wrong direction and it is surprising that it operates as well as it does, owing to the large amount of refrigeration thus introduced along with the fuel at the critical moment when all the heat and the highest temperatures possible

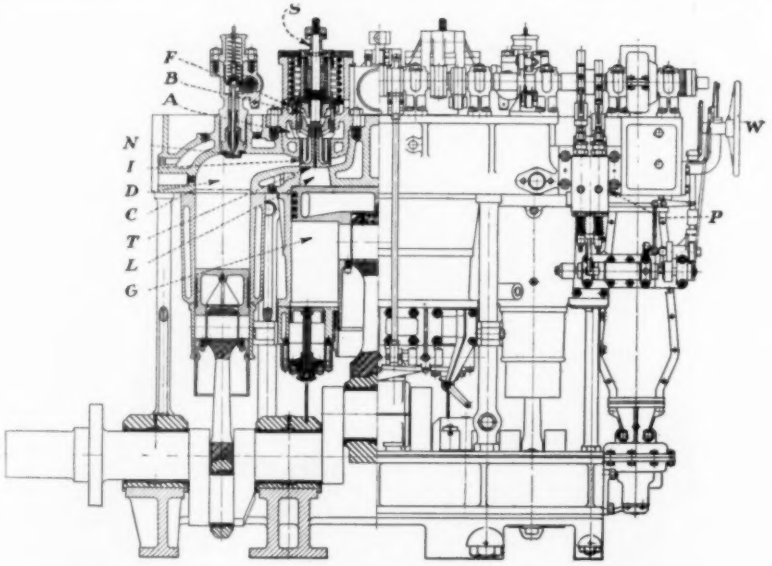


FIG. 14 SECTION AND ELEVATION OF COMPOUND OIL ENGINE, SHOWING CONSTRUCTION

should be fostered. This injection air represents roughly one-tenth of the air in the little combustion space and its adiabatic refrigeration factor per unit volume is almost exactly the same as the adiabatic heating of the combustion air. It is apparent that this is a very serious drawback. Moreover, the care of these pumps requires a high degree of skill, they deal with dangerous pressures when oil vapors are present, explosions in the intercoolers have proved fatal, and one United States builder stated recently that an excess of 11 per cent of the entire power of the engine is absorbed for driving these pumps.

46 As stated, solid injection only is used in the compound and the engine is controlled entirely by the fuel, viz., by lifting and timing the fuel valves. The constant-pressure trunk system is used with an accumulator in the line, and the pumps controlled thereby are adjusted at will. Two very small plungers are constantly acting.

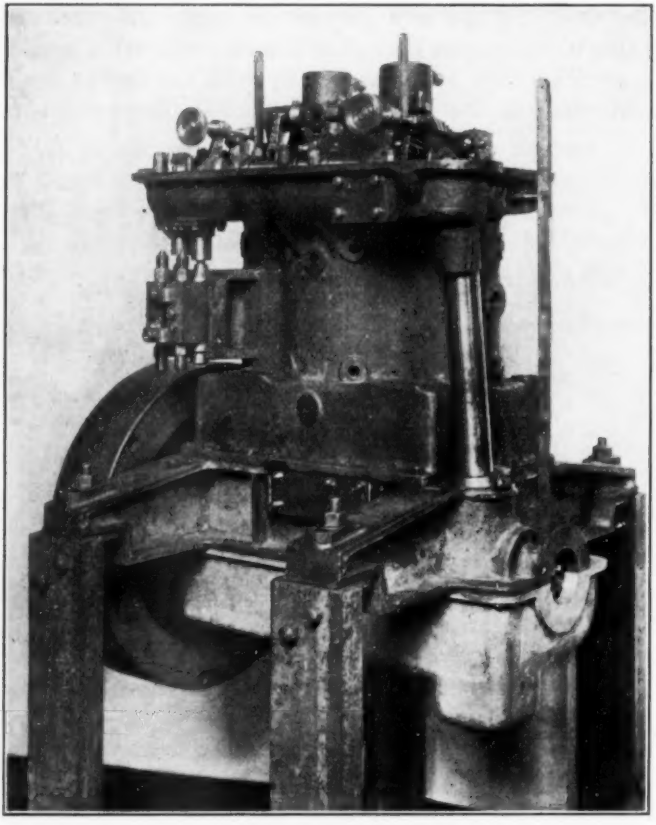


FIG. 15 SMALL 6:1 COMPOUND OIL ENGINE

The pistons, accumulator and the fuel valve stems are ground in and require no packing.

TYPES OF COMPOUND ENGINES

47 Fig. 15 shows an extremely small 6 : 1 compound oil engine with these two pumps in the left foreground, driven by eccentrics from the camshaft. The eccentrics act oppositely. The fuel-valve

cam levers are on eccentrics which regulate their dip onto the cam and, therefore, the quantity of fuel used. The two small upstanding levers control the eccentrics. The timing is effected by the little knurled hand wheels at the top by bodily moving the eccentrics tangentially to and from the cams. This cut also shows the relative size of the small boxlike air receiver located above the crankcase, operating between the two stages of the compression.

48 Fig. 16 shows the fifth compound engine of the series, a

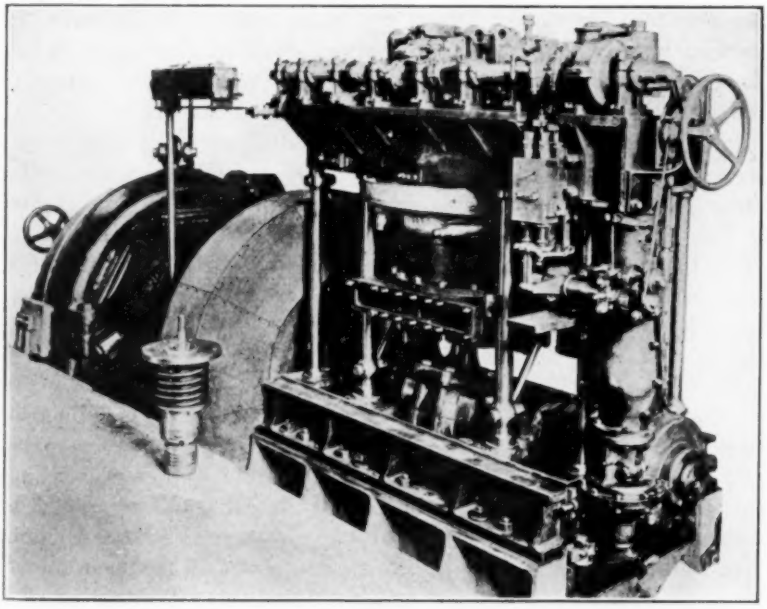


FIG. 16 10:1 MARINE-TYPE COMPOUND OIL ENGINE BUILT FOR HEAVY DUTY

10 : 1, built for heavy duty. Although this is a small marine type with high-pressure cylinders 7-in. by 11-in. running at 400 r.p.m. yet the size of the crank pitman-end in the lower center of the engine reveals the ruggedness of these parts. The fuel pumps are also shown here and the connection to the governor. The camshaft is on a shelf at the top of the engine to one side and is driven by skew gears. The electric generator forming the full load of this engine is shown in the background and one of the transfer valves with its bonnet cover stands on the floor in front of the engine. The comparatively

small size of the engine, although in the foreground, is notable and stands out in marked contrast with the engine of Fig. 17, the product of the largest Diesel builder. This is the standard marine engine for some seventy ships. The electric generator forming the full load of this engine (so marked) is also in the background, as in Fig. 16. The generators in the two cases of course being standard, give an unusually excellent basis of comparison. The compound works at a piston speed of about 700 feet per minute and its generator would be still smaller in comparison, should it be worked at the piston speed of about 900 feet per minute of the large engine. Incidentally, in the foreground of Fig. 17 there is also an excellent illustration of the three-stage, intercooled, high-compression air pump for spraying the fuel into the cylinders.

49 The large size and weight in the Diesels extends to all makes in somewhat different degrees. A line of Diesels made in the United States are reported to weigh 512 lb. per b.hp. The engine of Fig. 17 weighs 450 lb. per b.hp., and is of the crosshead type while the compound in Fig. 16 weighs less than 30 lb. per b.hp., the crosshead not having to be resorted to below four to six times the power of the simple engine.

INTERNAL CONSTRUCTION OF THE COMPOUND

50 Now as to the construction of the compound, Fig. 14 shows an elevation to the right of the center, and longitudinal section to the left, of the engine shown in Fig. 16. The two high-pressure or combustion pistons on their out stroke are at the ends, and in the center is the low-pressure at its extreme in stroke. The sturdy construction is indicated by the size of the crankshaft, about 50 per cent larger than in any other combustion engine of which the author has knowledge, approaching, as it does, the bore of the combustion cylinders themselves. The fuel pumps *P* and the control and manipulating wheel *W* are shown in elevation to the right. To the left the large dome of clearance *D*, forming the combustion chamber of the compound, stands out in marked contrast to standard Diesel practice, which is shown by the little space *C* between the solid horizontal line at the base of the clearance and the dotted horizontal line just above. The dome is large and forms an upward extension of the combustion cylinder, extending also to the right in a large sweep surrounding the transfer valve *T* which seals the transfer port *L*. The sleeve-like induction valve *I* is shown seated on top of the transfer valve and is controlled by the cam-operated fork *F*. The transfer valve and sleeve are lifted by a fork not shown, located in thimble *S*

near the top of the stem. The first-stage annular compression pump *G* surrounding the trunk piston below the low-pressure piston proper, delivers its air to a small receiver, which in turn discharges to the cored port *A* surrounding the induction sleeve *I*, the cooling action of which has been described. The little balancing cylinder *B* sus-

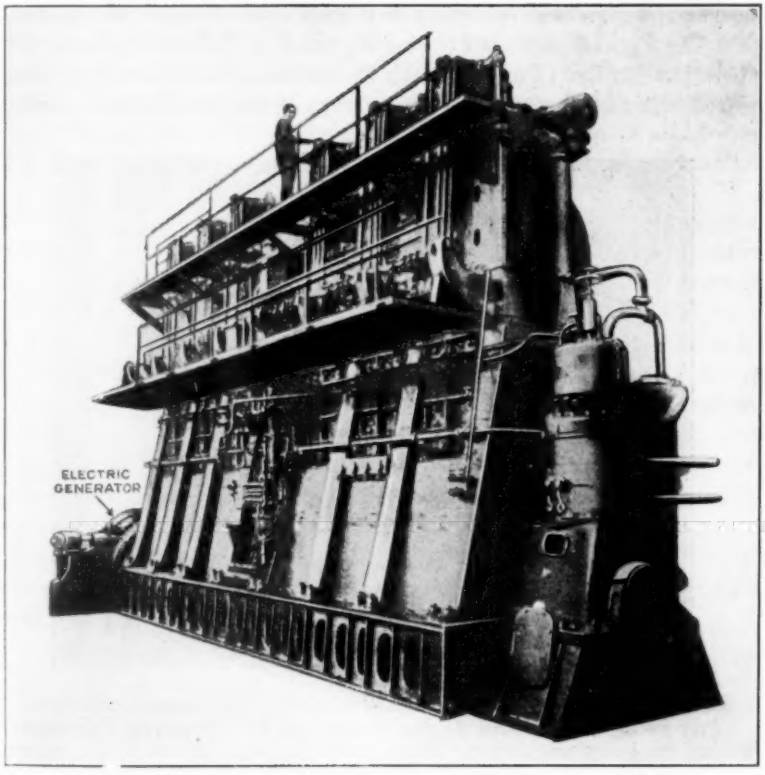


FIG. 17 STANDARD MARINE-TYPE DIESEL ENGINE

tains a permanent connection with the low-pressure cylinder. The solid-fuel injection valve and nozzle *N* are placed approximately over the center of gravity of the large masses of air in the clearance dome *D*.

51 The comparison with the large dome constituting the clearance and combustion space in this compound with the small distance between the solid horizontal line at the base of the combustion space

in Fig. 14 and the dotted line indicating the combustion space C in the normal Diesel, is simple to make and is significant.

52 It is understood that the two high-pressure cylinders are operating four-cycle, one 360 deg. back of the other, discharging alternately into the low-pressure, which therefore works two-cycle and delivers power on each down stroke. The cycle has been pointed out and the general operation will be apparent from this figure.

53 Fig. 18 shows an 8 : 1 compound of the same general construction as Figs. 14 and 16, with the cylinder heads raised, exposing the pistons together with the lower flange of the low-pressure cylinder.

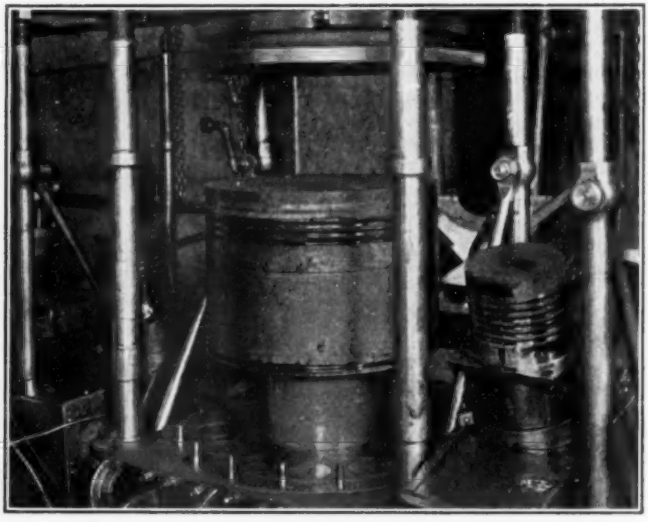


FIG. 18 COMPOUND ENGINE WITH CYLINDER HEADS RAISED, EXPOSING PISTONS TOGETHER WITH LOWER FLANGE OF LOW-PRESSURE CYLINDER

In the foreground stand the four columns plainly to be seen in Fig. 16. The high-pressure pistons are protruding up through their skirts on either side of the low-pressure, exhibiting their rings, the one to the left being partially hidden by a column. The low-pressure piston is elongated to take the entire slipper action away from the trunk, which is shown protruding below. Surrounding the trunk is the half-circle of air-induction valves of the first stage or supercharger, a similar row of eduction valves being out of sight on the opposite side. This photograph was taken after about a thousand hours run on this engine.

ORDINARY STANDARD CONSTRUCTION AND HIGH MECHANICAL EFFICIENCY

54 The construction is shown very completely in the figures; especially do the last two indicate the simplicity of the details, there being no part with which the ordinary engine builder is not perfectly familiar, nothing about the cylinders, pistons, rings or valves that lies outside ordinary good engine practice and construction. The radiating fins on the valves and the piston clearances and tolerances are the only points that vary to some small degree from the best Diesel practice.

55 A point of interest arises regarding the mechanical efficiencies of the compound. The two high-pressure cylinders in this case might be considered as constituting a two-cylinder Diesel. By separating them and adding one crank and the low-pressure piston between them, an increase of no less than seven times the power to the crankshaft has been realized in this engine. Pursuing this one step farther, to obtain this increase of power at the shaft, it is perfectly apparent that we have not added, nor are we driving, seven times the machinery. We are driving probably about twice the machinery, giving one apparent reason for the very low mechanical losses observed in this engine. These efficiencies have again secured an additional check by motoring the engine and taking a number of sets of cards, in several instances indicating as high as 93 per cent mechanical efficiency.

PORT VELOCITIES

56 In Fig. 14 the transfer port L to the low-pressure piston has only about one twenty-fourth of the area of the piston. The question naturally arises, what about wiredrawing through this port? This has been closely investigated and it is found that at 1000 ft. per min. piston speed, the velocities rise to 200 ft. per sec. only at a point near the greatest velocity of piston travel. This comes from the fact that by the time the center of the stroke is reached, where the velocities are highest, the expanding gases delivering the power find themselves very largely on the low-pressure side and therefore are not required to traverse the port. It is well known that no wiredrawing, as such, exists at the velocities named.

Fig. 19 shows a curve of the velocities through the transfer port at all points in the stroke of the low-pressure piston, on the basis of 1000 ft. per min. piston speed, 6 : 1 compound with a port having an area one twenty-fourth that of the low-pressure piston.

FIELD OF USEFULNESS OF THE NEW PRIME MOVER

57 To engineers versed in the problems of selecting and designing prime movers, the advantages of the compound combustion engine are readily apparent. Its light weight for a given power with resulting low first cost and capital charge, the low costs for foundations, the high speeds with consequent low costs for connected gen-

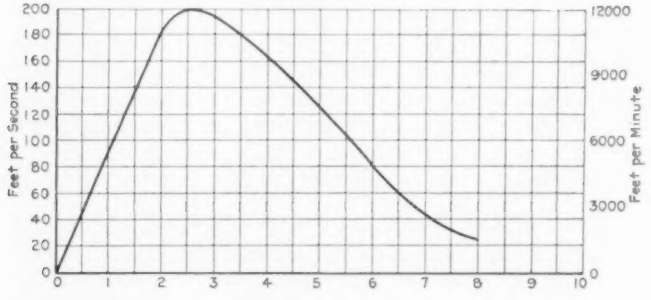


FIG. 19 CURVE OF GAS VELOCITIES THROUGH TRANSFER PORT AT ALL POINTS IN STROKE OF LOW-PRESSURE PISTON

erators, the small space required and the simplicity and economy of operation are important reasons for suggesting new fields for this prime mover, which has been proven of practical value in a long series of tests under working conditions. The compound engine is especially adapted for use for auxiliary or stand-by service in water-power stations or for carrying the peak loads in central stations where installations as large as 21,000 kva. have been contemplated. The important considerations are quick starting, non-deterioration of fuel reserve and flexibility of required fuel, amounting to almost com-complete independence of fuel quality. Other uses for the compound which readily suggest themselves are city water works and irrigation projects, either directly or by electrical distribution.

POSSIBILITIES OF THE COMPOUND DIESEL-ELECTRIC LOCOMOTIVE

58 It is not the purpose of this paper to discuss the complicated problem of Dieselizing locomotives, but the possibilities of the compound-Diesel-electric combination in this field should not be overlooked, as it presents all the well-recognized advantages of electric traction and at the same time includes several additional features, namely:

a It represents a component part of the central power station, with the advantage over the latter that it operates at the highest thermal efficiencies known — nearly three times the efficiency of even a very large power plant, i.e., three times the tractive effort is delivered to the rail from every pound of fuel burned.

b The actual cost of the delivery of each kilowatt-hour to the motors is far less than by any other known means. The remote station with its much lower thermal efficiency and the transmission losses, amortization and capital charges of the vast distribution system all add their burden to the cost of the kilowatt-hour finally delivered to the motors.

c The capital charges for main feeders, substations, bonded rails and shunted switches and frogs, and the third rail or elaborate overhead construction are eliminated. The tracks are used exactly as they now stand.

d It is estimated that the actual earnings of each Diesel-electric over the central-station electric, with all factors considered and performing the same service in both cases, will equal the first cost of the Diesel-electric each few months of service.

e The weight, distribution, wear and tear on the rails, ability to take short curves, and general flexibility of operation promise to better present steam-locomotive practice.

f The Diesel-electric being compounded and the expansion being carried on nearly to atmospheric pressure, there is *no noise at the exhaust* and only half the number of exhaust discharges.

g The cooling surfaces per unit volume of power gases are on the order of 40 per cent only, as compared with Diesel practice. Notwithstanding the fact that the engine secures ten times the expansion, yet the increased expansion deals with the gases at their lower temperatures and pressures, and in consequence fewer thermal units are passed to the jackets, power for power, than in standard or "simple" engines. All jackets are water-cooled with a single system; a circulating pump forms an integral part of the engine

detachable for inspection and cleaning. Large ducts are used to the radiator on top of the cab so that auto or convection currents continue long after the engine is shut down. Fig. 20 gives three views in outline of the compound oil-electric locomotive.

APPLICATION TO SHIP PROPULSION

59 We now come to the greatest present employment of oil engines, namely, their substitution for steam in ship propulsion. At the recent Petroleum Congress in Washington, the author was present when M. L. Requa, former Fuel Administrator of the United States, stated most emphatically that "steam equipment on board ships should from now on be regarded as obsolete." Dr. Norman, in Experimental Station Bulletin No. 19, quotes this and adds that "from the viewpoint of petroleum conservation and the fuel situation of the world, this statement cannot be too strongly endorsed."

60 The advantages of motorships are rapidly being recognized, and they are now being built in practically every maritime country more rapidly than ever before. The gross tonnage of ships equipped with Diesel machinery launched in 1913 was 60,000 tons. In 1919 the figure was 85,000, in 1920, 189,977, whereas there were building last year 454,502 tons of Diesel-driven ships, or over 7 per cent of the world's total under construction. In the light of the past ten years' experience the Diesel engine has proved an efficient, reliable and thoroughly seaworthy prime mover, suitable for a large proportion of the total sea-borne tonnage. In 1920, of 364 ships launched of more than 2000 gross tons each, 350, or 96 per cent of the total number, required less than 5500 i.hp. of machinery per ship — in numbers 96 per cent, and in tonnage 88 per cent, coming thus within the scope of the oil engine on its present basis.[1] It should be noted that in the last fiscal year ending June 30 steamships had increased 6 per cent, whereas motorships had increased 50 per cent.

NOVEL ELECTROMAGNETIC CLUTCH AFFORDS ELASTIC DRIVE

61 In connection with the development of the compound engine for marine purposes, and in order to provide any Diesel type of engine with speed flexibility equaling the reciprocating steam en-

[1] James Richardson in a paper on Recent Progress in Large Diesel Engines for the Merchantile Marine, read at the Conference of the Institution of Civil Engineers.

gine, there has been developed an electromagnetic clutch operating on an entirely new principle. This clutch furnishes a positive yet completely elastic drive brought about by a unique correlation of two oppositely applied forces, each producing powerful torques induced and brought under perfect control by the application of a very small amount of external electrical energy. The drive has no moving parts, and is rugged and simple in design and construction. It is small for its power and therefore capable of being built in practically any range of horsepower.

62 This new type of clutch transmits power entirely through air gaps and has no mechanical contact whatever between the driver and driven, eliminating wear and deterioration. It is capable of

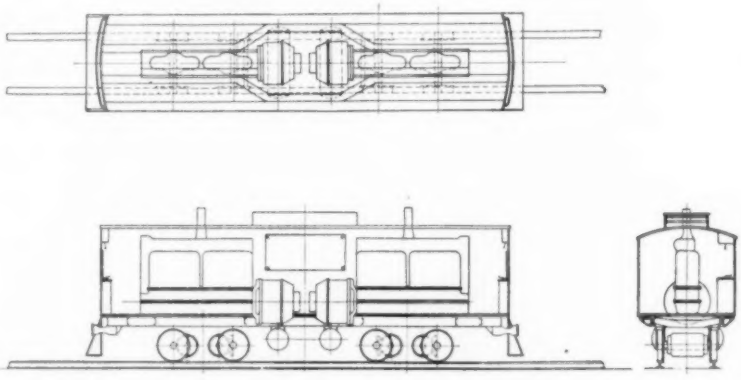


FIG. 20 PLAN AND ELEVATIONS OF COMPOUND OIL-ELECTRIC LOCOMOTIVE

remote control and can be operated at any speed from zero to full engine speed; the torque may be varied at will from maximum to minimum. The power required to operate the clutch at full load is but a small fraction of one per cent of the power transmitted. In one instance 525 hp. required 256 watts, or 0.065 per cent of the power transmitted. An outstanding feature of this clutch is that on direct drive or full speed it is magnetically locked, which insures perfect synchronism and no slip with an extreme increase in pull-out torque. At this and all times, in fact, the transmission has the "velvet touch" of an air drive and insures complete torque-wise isolation of the mass moments lying on the two sides of the clutch. As seen in Figs. 21 and 22, the clutch forms a part of even a small-sized flywheel of either a Diesel or compound with little, if any, alteration of its

mass moment. The figure shows the construction of one form of a complete embodiment and Fig. 23 gives the characteristic curve of its operation.

63 The clutch is characterized by two kinds of torque operating by opposite phenomena; the greater the differential or relative velocity between the driver and driven parts (indicated in Fig. 22), the greater the torque available for starting and for bringing up to synchronism. This phenomenon also provides for slipping and continuous operation at all fractional speeds by means of the loaded

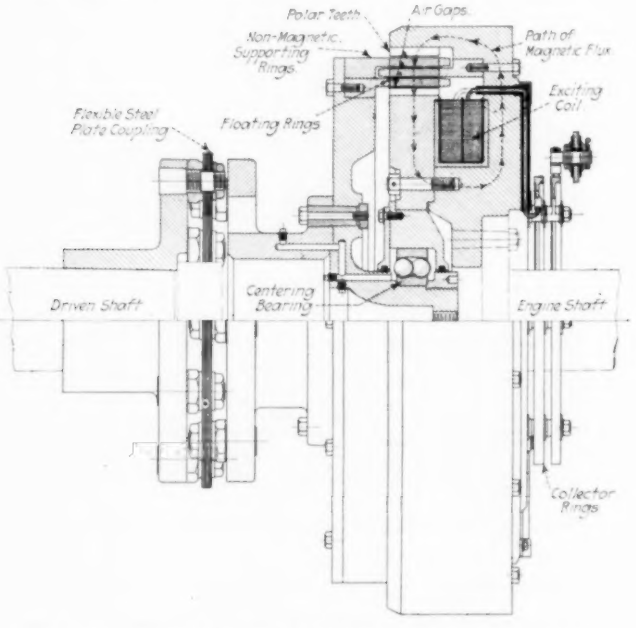


FIG. 21 AIR-GAP TYPE OF ELECTROMAGNETIC CLUTCH

secondary effect acting as an induction motor. The loaded secondary is indicated by the large light masses in the driven rings separating the blackened masses, which indicate the steel inserts.

64 The air-gap clutch drives through magnetic flux comprising four ring elements, two of which are driving and two of which are driven. The flywheel proper accommodates an exciting coil producing the magnetic flux shown in Fig. 22. At all fractional speeds or while this clutch is slipping, this magneto induction generates large currents in the driven elements which are now acting as short-

circuited secondaries, producing heavy drag torques under perfect control by varying the amount of coil excitation. However, when the speed comes up in the vicinity of synchronism, shown at the extreme left in Fig. 23, locking occurs and the parts assume the relative position shown to the right in Fig. 22, where enormous pull-out torques are present, as indicated by the height of the curve at the extreme left of Fig. 23.

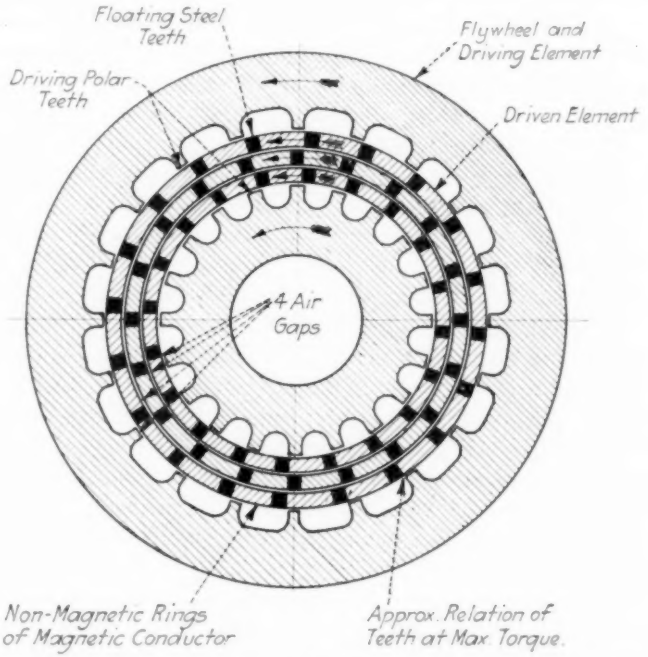


FIG. 22 SECTION THROUGH ELECTROMAGNETIC CLUTCH SHOWN IN FIG. 21

PROPOSED INSTALLATION OF PROPELLING MACHINERY

65 Fig. 24 shows in plan view a proposed installation of the propelling machinery, securing the advantage to both engine and tail shaft of entire freedom of each to run at its own best speed. A propeller for a given ship has a speed at which it gives its most efficient performance; likewise a combustion engine can be built most economically and to give its best performance for a given size if the designer is given entire freedom to choose the best engine speed, a condition which is extremely desirable.

MAGNETIC CLUTCHES VS. ELECTRIC DRIVE

66 The need for divided units has been satisfied by the adoption of the electric drive, gears with reciprocating engines having been found to thrash out and be entirely impractical. But though the electric drive is very heavy and the controls are decidedly extensive and intricate, and though they are all high in first cost and suffer double efficiency losses — generator losses and motor losses — yet the object of most efficient speeds for both engine and propeller is attained in a measure, all but a decided tendency to encourage high speed in both engines and propellers in order to cut down the generator and motor sizes, weights and cost. The electrical plant is of course cumbersome as compared to simple gears, but with reciprocating engines gears are the trouble, as stated; this has been recognized by engineers, but in view of engineering knowledge up to a very recent date, it seemed the most available solution.

67 In the case of the oil engine the greatest problem is simplified over the turbine in two ways: the speeds are lower and — the most important difference — the oil engine does not object in the slightest to full and quick reversing. Just here the new powerful electromagnetic clutch steps in with its air-gap drive and offers a solution by permitting complete latitude as to propeller-shaft speed regardless of the engine speed, the clutch being designed to insure any amount of fractional driving for indefinite periods, constituting a perfect complement to the oil engine and filling in any deficiency that the latter may have in maintaining extremely low crankshaft speeds.

68 The second major function of the clutch is that of allowing the pinion with its slight mass moment to become an entirely separate entity with all the advantage of complete and yielding flexibility both circumferentially and bodily, with no thrashing possible even when receiving the most powerful torques. Surprising as it may seem under these conditions, it finds itself completely free torque-wise, a result that has also been obtained mechanically with resultant elimination in gear-tooth wear. This at once makes comparatively inexpensive gears available, and the clutches are also low in cost and their efficiency exceedingly high, as pointed out above. This reduces the oil-engined ship propelling equipment to a few rugged parts and gives reliable performance at high efficiency.

ADVANTAGES OF THE MULTIPLE ENGINE AND AIR-GAP DRIVE

69 With this clutch and the engine combination shown in Fig. 24 several very important advantages are secured:

a There are available on a single propeller all the advantages and flexibility of a multiple engine equipment, where one engine may be shut down and completely disconnected for inspection, valve grinding, etc., and yet the ship be going forward at three-quarters its normal speed.

b Complete flexibility of the electric drive without the expense, weight and space of the electric generators and motors and the

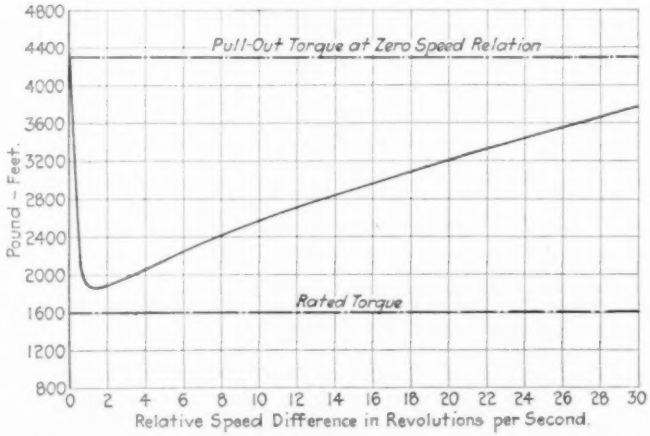


FIG. 23 SPEED-TORQUE CURVE OF ELECTROMAGNETIC CLUTCH

cumbersome electric control equipment for handling the heavy currents in maneuvering, and the double losses of generators and motors which are of substantial amount and a constant drag on plant and fuel economy.

c The simplest form of gear drive may be employed because the magnetic clutch allows the pinion to be a complete "floater." The pinion may thus accommodate itself to any want of precision and all sorts of idiosyncrasies of the main gear and teeth without shock. Any irregularities existing have to deal only with the small masses of the pinion itself and its stub shaft, being completely isolated from the large mass moments of the engine.

d The well-known irregularities in torque of any reciprocating engine and circumferential oscillation are entirely smoothed out by

the air gap in the clutch and are not permitted to reach the pinion; its specific tooth operation is thus completely safeguarded. The crash and general irregularities of the engine torques thus never reach the pinion and therefore can have no effect on the complete smoothness of its performance.

e The torque, being under complete control, can be lowered so as to safeguard the equipment against overloading, especially when sailing in obstructed harbors, near derelicts, and where floating obstacles are likely to be encountered by the propeller blades, thus providing an important emergency disconnecting gear breaking away from the large engine masses and allowing the propeller to "stop in its tracks" through the self-interruptibility of the magnetic clutch when reduced to fractional underload condition. In this way many disasters to the propelling machinery and interruptions to the service may be avoided.

f Nearly all revolving machinery is subject to periods, sometimes running into severe "criticals." These criticals always develop from the irregularities in torque of the mass moments within the engine pitted against outside mass moments aft. This can occur only when these are solidly coupled with each other, but if instead they are isolated, as by the cushion or air gap of a magnetic clutch of proper design, these troublesome criticals with their excessive stresses are completely suppressed and can never develop.

g The combustion engine is reversible and runs equally well in either direction. The magnetic clutch solves completely all maneuvering problems, and in this way stands out in bold contrast to the difficult maneuvering and reversing conditions introduced by the non-reversible turbine.

70 As seen in Fig. 24, a Kingsbury or equivalent thrust bearing may be located forward of the main gear. The pinions flanking the gear on either side are not necessarily diametrically disposed. This gives an excellent distribution of the machinery, compact engine-room arrangement, and ample space for the small oil-engine generating sets for the ship, which incidentally also supply the clutches with the very trifling amount of energy and any other auxiliaries outboard of the engines. The engines are completely reversing and self-air starting, and either engine may be completely isolated at will. When either engine is running, the other may be gradually or quickly started up by means of the clutch instead of by air, if desired. The engines under this arrangement run with extreme smoothness and under complete control. In one instance the clutch succeeded in

suppressing entirely a very serious "critical" which occurred in a 700-hp. submarine type Diesel near the normal running speed.[1]

71 After many years of development, under conditions at times none too encouraging, looking toward the introduction of this multi-stage principle and the material reduction in size and weight of these prime movers for ships, it will not be out of place, in closing, to ascertain to what extent this may be considered as having been accomplished to date, citing one or two specific instances.

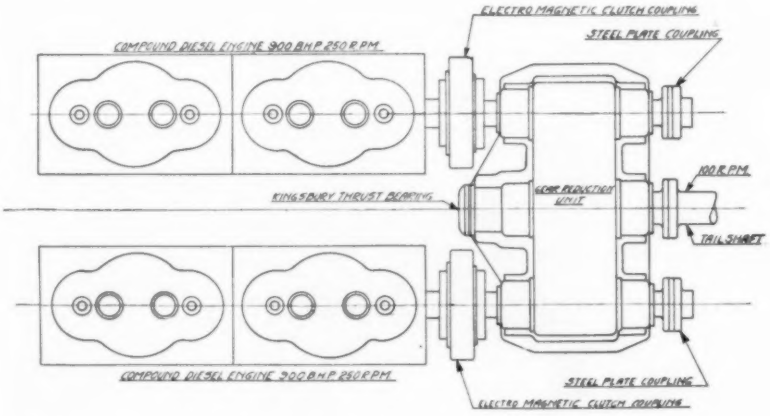


FIG. 24 PLAN OF PROPOSED INSTALLATION OF PROPELLING MACHINERY

SAVING IN WEIGHT AND SPACE OVER DIESELS

72 In Fig. 25 there is shown in side elevation to the left, and end elevation to the right, the comparison between the compound divided-unit geared drive delivering to the tail shaft the same power and speed as the standard Diesel unit of the largest manufacturer. The divided unit consists of two engines, each with four combustion cylinders, in two compound units. Note each of these engines gives the same turning moments and torque diagram as an eight-cylinder Diesel. In fact, with this divided unit we have eight combustion cylinders in the two engines with less than half of the cams, etc., of a six-cylinder Diesel, with all the advantages of two engines, including immunity from shutdown, possibility of inspection and other operating advantages which are well recognized. These two views serve graphically to give some idea of the saving in weight and space.

[1] See the *Sperryscope*, vol. 2, no. 9.

73 To aid in the comparison with the reciprocating steam engine, Fig. 26 has been developed, giving three views of the steam plant in shaded background. Against it there is out-lined the standard compound, two-engine, geared magnetic-clutch arrangement for the identical horsepower and speed at the tail shaft.

SPECIFIC COST COMPARISONS

74 A proposition worked out and submitted recently by a firm of naval architects is as follows: In a moderately high-speed tanker in the Atlantic service using about 4000 shaft hp. as compared with the ordinary Diesel such as has been specially cited above, there is a saving in weight of about 400 lb. to the horsepower, which would

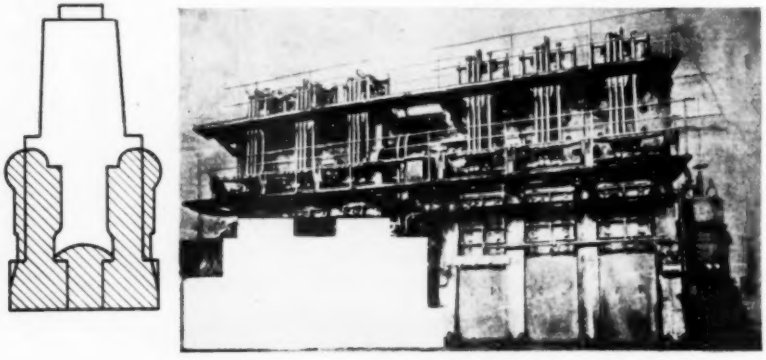


FIG. 25 COMPARISON BETWEEN COMPOUND DIVIDED-UNIT GEARED DRIVE AND STANDARD DIESEL ENGINE DELIVERING TO TAIL SHAFT THE SAME POWER AND SPEED

mean an extra earning capacity for the tanker of $120,000 a year. The comparison with a steam-driven tanker is still more favorable, because in addition there is the earning capacity due to the extra saving in tonnage of fuel oil required as compared with that used under the boilers, giving another item of $134,000 a year, added to which is $85,000 saving in the fuel item itself, making in all $220,000 to be added to the $120,000. This makes $340,000 or not far from the cost of the entire equipment, or a 100 per cent dividend annually. The saving of personnel and subsistance, amounting to about $30,000 is an additional saving over steam.

75 As to the turbine-electric drive, a comparison with a recent installation of this type of 3000 shaft hp. for a 17-knot ship gives the following results:

Steam-electric propelling machinery		670 long tons
Weight per shaft horsepower		500 lb.
Weight Sperry compound and auxiliaries:		
2 lighting sets, etc.	70 tons	
Gearing and electric clutches for same propeller speed	15 tons	
		85 long tons
Tonnage saved and available for cargo		585 long tons
Weight per shaft horsepower of entire plant		63 lb.

There is also a saving of space available for cargo of 28,590 cu. ft. in favor of the compound geared drive with electric clutches.

76 These figures speak for themselves. They show, as compared with the compound combustion engines employed in two units

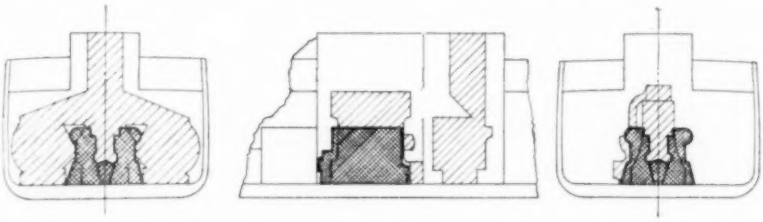


FIG. 26 COMPARISON BETWEEN RECIPROCATING STEAM PLANT AND GEARED COMPOUND DIESEL ENGINE

as herein proposed, that this ship is using in addition to 350 to 400 per cent too much fuel,

240 per cent too much space for her propelling machinery, and
790 per cent too much weight for her propelling machinery,

besides having to rely upon one engine, one electrical generator, one motor and one set of control gear for reliability of operation.

CONCLUSION

77 In combustion engines the high-pressure principle is scientifically correct, having brought to the prime mover its choicest heritage, as stated, viz., the highest thermodynamic efficiency known. Broadly speaking, however, these various engines represent only the early stages of the development. The best of present embodiments in simple Diesels, though much refined as to certain details, are, generally speaking, extremely crude, their design is based on such a ridiculously small quantity of air per charge, and their material efficiency is too low to be tolerated. Further, very large structures are required to produce small powers.

78 This paper presents the results of research extending over a series of years, looking to the correction of these and other serious faults and to very greatly increasing the material efficiency. This is found to bring with it a number of distinct advantages and, it is believed, marks a definite advance. It is high time that this next logical step be taken so that the full capabilities of the high-pressure principle in combustion engines may be realized in service by the adoption of modern multi-stage methods. Professor Watkinson is correct when he says: "Only by compounding, as in the case of steam, are we to make the next great advance." The author's contribution toward this most desirable end has been set forth in the preceding paragraphs, and some of the more important results reached have been outlined. It has been the aim to state both in simple terms, free from too much technicality. The thermodynamic calculations, heat analyses and other technical features of the work are, however, vitally interesting and may be collected and presented at some future time.

79 Any contribution toward increasing the availability of combustion engines will not fail of appreciation by the engineering and prime-mover interests generally, including the marine. The high-pressure type presents the true line of attack, and contributions in this line will aid in lifting embodiments of this most useful principle to the high place it richly deserves. It is believed that the results for the first time announced in this paper warrant the prediction that engines employing the multi-stage or compound principle will occupy a dignified place in the combustion-engine art. If this principle should assume leadership, it will of course only be because of its demonstrated practical usefulness and general merit. At present these appear to be of such magnitude as to at least insure wide adoption.

80 This paper would not be complete without acknowledgment of the able assistance the author has received in conducting this work from his engineers, headed by Mr. Herman J. Scharnagel, formerly civilian Diesel engineer with the U. S. Navy.

DISCUSSION

GEORGE A. ORROK. The Society is to be congratulated on securing such a paper as Dr. Sperry's, which is the first presentation of his work on compound Diesel engines.

While the inception of the paper does not go back to the early days of the combustion engine, the author's work, starting as it does in 1892 and continuing to the present time, represents nearly a third of a century of development work. Notwithstanding the work of Diesel, Clerk, Junkers, Ricardo and others, the Sperry engine appears to have solved the problem. The transfer valve between the high- and low-pressure cylinders, which has been the cause of the greatest amount of trouble, has apparently been put into workable shape. The writer saw this transfer valve which had been working in the engine for over 2000 hours, and as far as he could see there was no deformation and no evidence that it had ever been exposed to any serious degree of heat. The surfaces were good. The cylinders looked very much like steam-engine cylinders that had received good care and not at all like gas-engine cylinders.

The author's design lends itself to a reasonable amount of supercharging, which increases the power of the engine. Too much stress cannot be laid on the mechanical atomizing of the fuel which, in the writer's opinion, cannot be obtained by an air jet and must be done mechanically with preheating of the fuel to a proper point. It has already been learned in burning fuel oil under a boiler that the oil must not be atomized either with steam or compressed air, and that the only way to burn it with high efficiency is to put the pressure behind it and heat the oil to a point where its viscosity is such that it will truly atomize. While opinions may differ as to the gains due to the detonation of the atomized fuel, it is at least plain that under the conditions obtaining in the author's engine the practice has not been attended with the customary evil results. It is a curious commentary on the art of designing internal combustion engines that the 200-hp. Sperry engine which has run under load 3000 hours should follow in a very large measure the canons of steam-engine design and not to any great extent the newer and much more strenuous rules used by modern Diesel-engine designers.

FRANCIS HODGKINSON. None of us can fail to be interested in this design of a compound Diesel engine. By virtue of compounding, the compression space in the high-pressure cylinder occupies a

greater proportion of the stroke. It is therefore changed from the thin lamination such as Mr. Sperry has shown to obtain in the case of the Junker type of engine to one of better form, one more nearly resembling a volume having the least surface. The writer would inquire if Mr. Sperry does not think that this is the principal cause for his having secured detonation. This should, the writer believes, add to the efficiency of the engine.

The idea of compounding an internal combustion engine is, of course, not new, the stumbling block having been the transfer valves for passing gases at the right time from the high-pressure to the low-pressure cylinder. A former teacher forcibly inculcated in the writer the principle that when undertaking the design of a piece of machinery, he should endeavor to make the stumbling block, or the difficult phase of the project, the feature of success. The writer thinks Mr. Sperry has followed this principle, for he has seen this valve after many hours of operation in the experimental engine he has built and has marvelled at the fact that it shows no signs of burning or distress of any kind. Plainly, great ingenuity and skill have been bestowed upon the design of this valve. It is more likely to meet with success than any device hitherto proposed for the purpose. Its real practicability can only be determined by extended use in larger sizes.

It would seem that the whole cylinder structure is complicated on account of the ports to this transfer valve. They do not have the extreme simplicity of the Junker engine and those of similar type. Perhaps in larger sizes, cracking of castings may be experienced from this cause.

Charles D. Mosher. The writer would like to ask if, due to the excessive pressures, the limitations of size of the engines will not be reached, by the necessarily great thickness of cylinder walls, before anything like large powers per cylinder are obtained? Also, in firing the charge of a cylinder of large volume, will the propagation of flame be sufficiently rapid to penetrate through the entire mass, at a high number of revolutions, and result in efficient and complete combustion before the exhaust valve opens? If this is the case, it seems that the smaller engines may have a much higher efficiency than large engines. In other words, the engines may reach a size where the efficiency may fall off very considerably.

C. C. Trump. The writer would like to ask how the life of the fuel valve compares with the life of other Diesel-engine valves.

The author explained how the transfer valve was kept cool, but if the oil is heated going through the fuel valve in order to make a spray, it seems as though there would be difficulty in keeping the fuel valve cool and insuring for it a long life. The writer would also be interested in the mechanical and the thermal efficiency of the engine as compared with other machines.

W. J. WOHLENBERG. Several questions concerning the efficiency of the Sperry engine have been asked. The two conditions which have the greatest influence on efficiency are, of course, first, the rapidity and completeness of combustion and, secondly, the expansion ratio.

The Sperry engine seems completely to meet the first condition and is a step in advance of the ordinary Otto engine in meeting the second condition. It is obvious that the cycle on which this engine operates is fundamentally an Otto cycle in which the compression pressure has been increased. This should result in a gain of efficiency if other conditions do not operate against it. The higher compression pressure as it is obtained in this cycle must, it seems to the writer, result in exceedingly high temperatures with correspondingly greater jacket losses.

Keeping in mind the second condition for high efficiency, the writer would like to propose, for purposes of comparison, an intercooling cycle which he devised several years ago and which if compared with the Otto cycle, assuming perfect combustion for both cases, leads to important conclusions concerning efficiency.

An analysis of the theoretical fuel consumption for the two cases results in figures which are considerably in favor of the intercooling cycle providing the intercooling cycle has been the means of increasing the possible compression pressure. If the compression pressure in the Otto cycle is equal to that existing in the low-pressure cylinder of the intercooling cycle the fuel consumption figures will be from fifty to sixty per cent in favor of the intercooling cycle. If the compression pressure in the Otto cycle is equal to that in the high-pressure cylinder of the intercooling cycle there may still be a slight advantage for the intercooling cycle under certain load conditions. This in spite of the fact that work done is rejected as heat during the constant-pressure cooling process. This loss is more than made up for because, by intercooling, the volume of the charge after compression is less in the intercooling cycle. The physical conditions determining this difference are more clearly

seen by referring to Fig. 27, in which the net work of the intercooling cycle is represented by the area *BCDEFGH*.

We will consider a compound engine with an intercooler between the cylinders. Point *B* represents the condition of the charge in the low-pressure cylinder at completion of the suction stroke. From *B* to *C* the charge is compressed in this cylinder, followed by discharge at constant pressure into the intercooler. In this process the volume of the charge is decreased from that at point *C* to that at point *D*. The intake valve of the high-pressure cylinder is now opened and the charge at constant pressure and

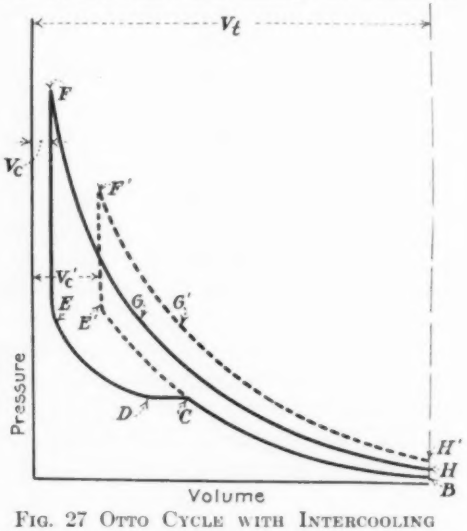


FIG. 27 OTTO CYCLE WITH INTERCOOLING

comparatively low temperature is introduced into the high-pressure cylinder as the high-pressure piston recedes.

As the piston returns, the high-pressure, low-temperature charge is compressed along a line *DE*. Ignition occurs at *E* with the consequent pressure rise *EF* during the combustion period. The charge now expands in the high-pressure cylinder from *F* to *G*, at which point the transfer valve opens and expansion is continued down to the point *H* in the low-pressure cylinder.

In this figure the net work of the cycle on which the Sperry engine operates is represented by the area *BCE' F' G' H'*. The dotted portion shows how the Sperry cycle differs from the intercooling cycle. Corresponding events are indicated by the same letters but with the index prime for the Sperry engine cycle, when the charge

condition differs in the two cases. By inspection of the figures the following facts are obvious.

1 For the same volume in the low-pressure cylinders and equal compression pressures in the high-pressure cylinders the expansion ratio of the intercooling cycle will be much greater than it is for the Sperry engine cycle, i.e.,

$$\frac{V_t}{V_c} > \frac{V_t}{V'_c}$$

2 For the same fuel content per charge and the same compression pressure at ignition the pressure rise during the combustion period will be much greater for the intercooling cycle than it will be for the Sperry engine cycle; or for the *same temperature rise* during the combustion period the pressure rise *EF* for the intercooling cycle will be higher than the corresponding pressure rise *E' F'* of the Sperry engine cycle.

3 Because of the greater expansion ratio for the intercooling cycle the final pressure at exhaust *H* will be considerably lower for the intercooling cycle than it is at exhaust *H'* for the Sperry engine cycle.

4 For the same temperature rise during the combustion periods the final temperature at *F* will be lower than the final temperature at *F'*. The latter condition will tend to result in lower jacket losses during the combustion and expansion periods.

5 The following physical conditions are in favor of the intercooling cycle:

a Higher pressure and smaller volume of charge at completion of combustion with consequent greater expansion ratio and smaller exhaust pressures and losses

b Lower average temperature of the charge during the process, resulting in lower jacket losses.

6 To counteract these apparent gains the conditions for good combustion may be unfavorably affected by the lower temperature at the beginning of combustion. This will, however, at least partially be compensated for by the smaller volume of charge. Where liquid fuels are used the amount of intercooling must obviously be so regulated that the dew point is not reached during the compression which follows.

7 It is now an obvious and also an important fact that preheating of charges increases the efficiency only in so far as it promotes

better combustion. In other words, the ratio of the available energy which is converted to work during the expansion stroke is greatest for a given maximum pressure when the maximum temperature is lowest. Intercooling affects this condition favorably and preheating unfavorably.

The last statement is of particular importance when considering the value of reclaiming heat from the transfer valve by the incoming charge as is done in the Sperry engine, it being quite evident in view of the above, that any heat so reclaimed can only decrease the inherent efficiency of the cycle unless it promotes better combustion. It is probably a necessary evil to keep the valve cool and this must be accomplished in spite of the resulting loss in efficiency because of it.

J. C. SHAW. Mr. Sperry's engine is interesting, as well as ingenious, particularly in the methods employed in effecting the idea of compounding the oil engine, and he is to be commended for his research along the lines indicated.

It would appear on casual observation that there might be some saving in weight over the simple engine, due to the low-pressure element of the compound engine having twice as many working strokes as the corresponding part of the four-cycle simple engine. This advantage, however, is evidently more apparent than real, as will be shown later, provided the same general construction is adhered to for either engine. A similar argument has been frequently advanced for the two-cycle engine over the four-cycle for effecting weight saving, but which has not always proved correct in actual service conditions.

The claim for inherent superior economy of the compound engine over the simple engine is to be seriously questioned, and it is particularly to be regretted that the author has failed to supply any actual figures for fuel consumption to substantiate this general claim. The parallels that have been drawn with other types of machinery, as reasons for anticipating the better efficiency, are incorrect as a rule, and more or less misleading.

The compounding of the steam engine is cited, but this compounding (as is well understood) is done entirely for another purpose, non-existent in the gas or oil engine, that is to prevent the condensation and reëvaporization of the steam in the cylinders. In the oil engine the gas medium is approximately ideal and is not disposed to condense in the cylinder.

The author similarly makes mention of the large expansion in a steam turbine as a reason for greater expansion desirable in the oil engine, but the comparison is again incorrect. The turbine must work down to a very low absolute pressure, corresponding to the modern high vacuna obtained in condensers, and the final specific volumes handled by the turbine are accordingly very large. The turbine, furthermore, is not handicapped (as is the oil engine, or any reciprocating engine) by having restricted exhaust openings. In the oil engine the back pressure cannot be less than atmospheric pressure, large expansion ratios not being required, and there must be some "toe loss" in card area, or slight pressure drop, so as to get the gases through the exhaust valves, which are limited as to size, as previously stated.

The author also cites the multi-staging of the air compressor as a parallel reason for doing the same in an oil engine. It may be an aid from a lubrication standpoint for the compressor, as stated by the author, but it is primarily done for another reason, as is well known, and which is to permit of intercooling the air to reduce the amount of external work required to be done on the air. Any loss of heat in oil engines, either during expansion or compression, beyond keeping the temperatures within practical limits, conversely is to be avoided.

The author further advances the claim for superior economy by "hanging on to the pressure in the gases throughout two strokes, clear across the card twice, thus extracting much more of the power they contain." This is difficult to comprehend, as the pressure in either the compound or simple engine should follow the same general adiabatic curve of expansion, due to the burnt gases being composed of the same constituents of combustion. In the formula PV^{γ} = Constant, for adiabatic compression or expansion, the exponent γ for air is 1.403, and for exhaust gases is usually taken as 1.37. To sustain or "hang on to" the pressure more during expansion, as suggested, would have the effect of lowering the exponent below 1.37 and to approach the isothermal curve in which the exponent becomes unity. This would be undesirable, as it would indicate after burning and loss in efficiency. This efficiency loss is clearly indicated by the theoretical thermodynamic formula for thermal efficiency, which for the constant pressure engine is

$$E = 1 - \frac{1}{R^{\gamma-1}}$$

and for the constant volume engine is

$$E = 1 - \frac{1}{R^{\gamma-1}} \times \frac{R^{\gamma-1}}{\gamma(R^{-1})}$$

in which r = the ratio of clearance volume to total volume at end of stroke, and R = "cut off" or ratio of volume where fuel is stopped being admitted to that of clearance volume. It will be seen from the above formula that the efficiency varies directly with γ, and that the pressure should fall rapidly enough to approach ideal adiabatic expansion. It also follows that with the adiabatic expansion less expansion ratio in cylinder is required, and should not be carried beyond what is practically necessary. The high-speed engine, especially working on the constant-pressure cycle, is prone to after burning and raising the end of the expansion curve, and the back pressure

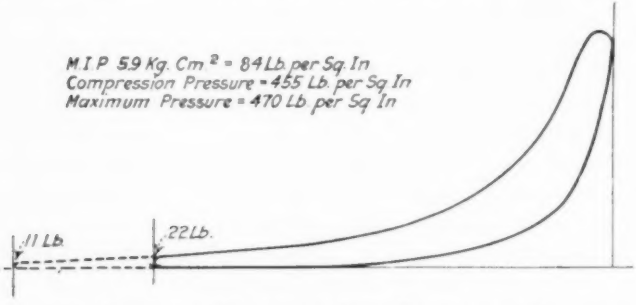


FIG. 28 INDICATOR CARD FROM PORT ENGINE OF MOTORSHIP *William Penn*

is also augmented by the choke in the exhaust valves, in itself due to the high speed. On the other hand, in the slow-speed engine the burning is much more complete in the beginning of the stroke, due to the time element being longer, and there is less restriction in the exhaust passages than with the high-speed engine.

The author shows in Fig. 2 a theoretical card of a two-cycle Diesel engine with exaggerated "toe loss," and indicates in extended dotted lines the additional area which is supposed to become available with the compound effect. In contrast to this card there is produced in Fig. 28 a tracing of an actual card taken from the port engine of the motorship *William Penn*. The cylinder size is $29\frac{1}{8}$ in. diameter by $45\frac{1}{4}$ in. stroke, and the M.I.P. of the card shown is 5.9 kg. per sq. cm., corresponding very closely to the normal rated power of the engine when turning at 115 r.p.m. The end of the expansion curve is about $1\frac{1}{2}$ kg. per sq. cm. or 22 lb., which has also

been checked by calculation from other points on the expansion curve. This expansion curve has been extended in dotted lines by calculation to half the above actual release pressure, should it be possible to work to the lower pressure, and the additional area that becomes available is found to be less than 6 per cent. This would necessitate increasing the expansion ratio by about 27 per cent with the additional weight and friction of parts involved, and it is accordingly extremely unlikely that any improvement in efficiency could actually be realized with the greater ratio.

It should be stated here, by way of interest, that in the new design of Burmeister and Wain engine of unusual slow speed of revolutions, especially adapted for single-screw vessels, the thermal efficiency has been increased by from 5 to 6 per cent. This is due chiefly to the better burning effect at the early part of stroke, and hence being able to approach more nearly the ideal Diesel card.

It is noted that the author has experimented with both the constant-volume cycle and the constant-pressure cycle, and has abandoned the former in favor of the latter, using the solid injection. This is not surprising, as it is generally known that, with the small high-speed engines, the solid injection works best when operating on the explosion or detonating principle.

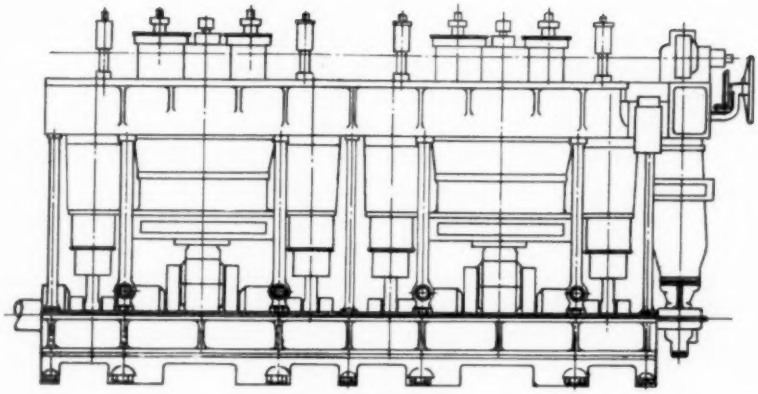
An actual card is given in Fig. 12 for the 7×11-in. engine, working on the explosion cycle, and it is unfortunate that no card scale was given for checking the actual pressures. By comparing it with the constant-pressure card in Fig. 11, it is assumed that the scale is about 360 lb. to the inch, as usually employed. On this assumption the maximum pressure would be about 600 lb., and the compression pressure about half this, or 300 lb., which latter is sufficient for automatic ignition with solid injection. The back-pressure line of the high-pressure exhaust as well as the air charging line is also shown on this card, but it does not follow, as one is led to assume, that the corresponding pressures registered outside the cylinder will be the same. It would have proved very interesting had corresponding actual cards been taken from the low-pressure cylinder, top and bottom, and had the pressures so obtained been referred to the low-pressure cylinder, as in steam practice. This would show the actual card area losses due to pressure drops between the various cylinders, as is to be expected. It is to be noted from the high-pressure card given that the negative work of the supercharging side of the low-pressure is actually about equal to the work done by the working side, taking into consideration their relative areas as

shown in engine section, Fig. 14. It would appear that at small fractional powers there is danger of overexpansion, and of the low-pressures doing very little work and acting as a drag on the system. The engine differs from a compound steam engine in not being able to vary the "cut off" of the low-pressure for equalizing the powers and expansions between cylinders to suit the actual conditions.

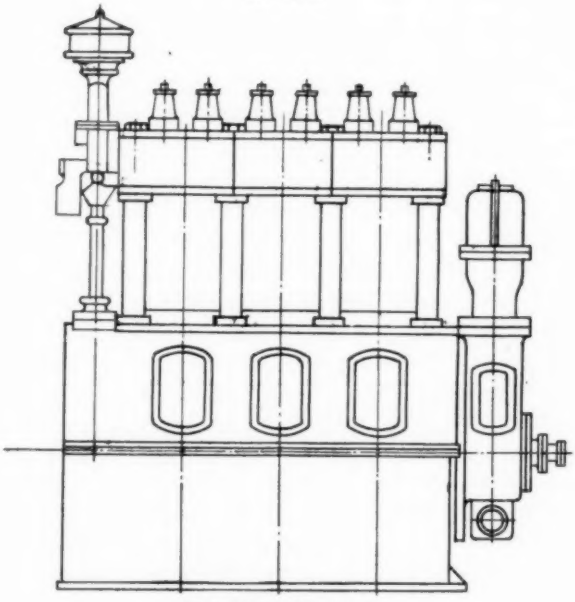
The author, however, has furnished us with some ideal cards shown to small scale, Figs. 3 to 8. Comparing cards 5 and 6, showing high-pressure cylinder theoretically functioning both as constant-volume and constant-pressure cycle with same compression pressure, it is seen that the compression pressure is about 650 lb., and the explosion pressure for the constant-volume cycle is over 1000 lb. This is estimated from point B, given as 113 lb. The increased theoretical efficiency of the explosion card is readily understood from the theoretical formulae, previously given, and is on account of the greater card area available in common with the high maximum pressure employed. It is seriously doubted, however, if the theoretical card could be anywhere nearly approached in practice, especially in the larger commercial sizes. This is on account of the high shocks and strains that would be produced and the heat troubles encountered within the high-pressure cylinder and transfer valve, but not so evident in the small experimental engine of small dimensions.

Mr. Sperry strongly advocates his high-pressure explosion engine for cargo-ship propulsion for replacing the present well tried slow-speed direct-drive engine. The latter, though acknowledged by him as being responsible for the present high standard of the motorship, he particularly criticises for its excessive weight. In so doing he has overlooked completely the shortcomings of the explosion engine and high-speed engine. The explosion engine is inherently unsuitable for marine drive, due to the shocks at dead center and lack of flexibility at reduced powers. The slow-burning engine, on the other hand, is specially adapted by lack of shock and by great flexibility, due to having the burning take place after the piston passes dead center, without any appreciable rise in pressure.

To prove his point for saving in weight the author mentions a mean effective referred pressure of 300 for the high-pressure cylinders. This has been doubled or referred to one high-pressure cylinder only and increased to 660 lb., and it is pointed out that this is ten times that usually employed in the simple engine. From this he infers the weights, respectively, should be inversely in the



Sperry Marine Engine
400 r.p.m.



B. & W. Auxiliary Engine
400 r.p.m.

FIG. 29 COMPARISON OF SIZE OF SPERRY AND B. & W. OIL ENGINES

same proportion. The comparison of mean pressures would be more nearly correct had that of the compound been referred to the low-pressure cylinder, and which in the case of 1 to 10 engine, would

have brought it down to that of the simple engine. The particular design of engine as advocated for marine work is shown in section, Fig. 14, and in the photograph, Fig. 10. It is noted to be of very light skeleton construction, and of the trunk piston type, and is accordingly entirely unsuitable for the particular class of service. The comparison of this high-speed engine on a weight basis, without taking into consideration the additional weights of the complicated clutches and gears also involved, with that of the slow-speed direct-connected crosshead engine is considered very unfair. An attempt has been made by the writer to arrive at a true comparison between the particular compound engine and a simple engine of similar construction. In a contemporary copy of the Sperry Company's publication, known as the *Sperryscope*, there is shown in outline an engine having the same general over-all dimensions as shown in Fig. 14, and designed for the same revolutions of 400. From this it can be assumed that the engines are one and the same. The engine illustrated in the publication is direct-connected to a 50-kw. generator, and the statement is made that are two engines placed end on end is suitable for driving a 100-kw. generator. Acting on the above clue and suggestion made, a tandem compound engine, shown in Fig. 29, has been laid down from Fig. 14, and to the same scale has been drawn a Burmeister and Wain auxiliary engine, designed for the same revolutions, and for direct connection to same size generator. The Burmeister and Wain engine, which is a new type, designed with the idea of installing in limited spaces, is four-cycle, totally enclosed, and has three cylinders of $11\frac{1}{16}$ in. diameter by $13\frac{3}{4}$ in. stroke. The diameter of the low-pressure cylinders of the Sperry engine is figured from the 1 to 10 ratio as 22 in., and the high-pressure cylinders are 7 in., as given, by 11 in. stroke.

The simple engine is seen to have about 55 per cent less cubic displacement than the compound, and which figure does not take into consideration the supercharging displacements of the low-pressure cylinders, but which are essential accessories to the high-pressure cylinders. There are three working cylinders as against six, and one set of sizes of parts instead of the two sets required for the compound arrangement. The compound engine, furthermore, has a larger number of objectionable automatic suction valves, inaccessibly located, and the trunk type of air piston is a type of construction to be avoided, having been abandoned by most two-cycle builders as unsatisfactory in operation.

It has never been the intention of these Danish builders who

have had ten years' experience in building motorships, and are chiefly responsible for the large number of successful ones in operation, to use the auxiliary engine construction for the main drive by means of gears or other means for reducing the revolutions to that of the screws. Their experience shows that reliability is of the utmost importance for the main drive, and this can be best obtained only by the slow-speed crosshead engine, working on the pure Diesel cycle, and designed for direct coupling to propeller.

The weight of the standard Burmeister and Wain engine, including attached piping, grating and flywheel, and as heretofore built, is 350 lb. per s.hp., and not 450 lb., referred to by the author in error, and it should be added here that, by a newer construction, the weight is actually being reduced by 10 to 15 per cent below the 350 lb.

The actual fuel oil consumption on the above standard Burmeister and Wain engine, as used with twin-screw vessels, is from 0.38 to 0.39 lb. per b.hp. With oil of 19,000 B.t.u., and with the new special slow-speed engine, adapted to single-screw vessels, the consumption is about 0.02 lb. lower. It is thought that these figures will stand comparison with any that may have been obtained with the compound engine, though not given in the paper.

In Fig. 17 is shown by the author a 1550-i.hp. Burmeister and Wain marine Diesel engine, coupled to a generator in the background, and which appears small due to the effect of perspective. The author states that the generator so shown, (but used for test purposes only), "forms the full load of the engine." He has been misinformed, as the generator in question is used for absorbing only the power of three cylinders at a time of the particular engine, with the other three cylinders acting as additional load.

Fig. 25, it should be explained, is a picture of the port engine of the motorship *William Penn*, owned by the U.S. Shipping Board, and now circumnavigating the globe on her maiden voyage. Embodying as it does the accumulative experience over a large number of years of the most successful marine builder, and in view of the inconsistencies of the author's claims, as pointed out, it stands to reason that the comparison of this engine with the hypothetical one of the author can have no particular value bearing on the subject.

It would have been of more good to the engineering fraternity had Mr. Sperry restricted himself to the true merits of his engine, giving the complete test figures, instead of creating possible confusion by inaccurate comparisons and generalities, particularly in

reference to its doubtful application to marine use. He is to be commended, however, for his endeavors, and it is hoped that the members of the Society will be favored with more complete data at an early date.

THE AUTHOR. Mr. Orrok and Mr. Hodgkinson have touched on a very vital point. The transfer valve has been a stumbling block ever since the first compound was conceived, and everyone has realized it, except Junker, who must have "slipped a cog" when he tried to uncover a valve at about 2700 deg. fahr. temperature. But such are the temperatures we must cope with.

The air injection of fuel mentioned by Mr. Orrok is most interesting. If air is compressed to the point required in the Diesel cycle, we get a temperature of about 1200 deg. If this air is compressed again to another 500 lb., which is the regular pressure of the air when it is used, there results another great increase in temperature. When this air is injected into the cylinder, that temperature drops back again, creating a tremendous refrigerator effect just where a refrigerator cannot be allowed. The incandescent air must not be refrigerated; all the heat must be left in it. It ought to be guarded still further, because it has been calculated that the amount of heat taken up in the vaporization of the fuel is 80 per cent of the heat in the air.

The curse of the Diesel today is the tremendous capital charge. Here are two ships fitted out in Philadelphia, the same exactly, one with steam and the other with Diesel engines. The Diesel costs $306,000 more than her steam competitor. Diesel engines weigh enough, one might say, to sink the ship. There is less space on a Diesel-engined ship by just this amount, and yet, the Diesel engine has such an advantage in fuel economy over its steam competitor that on an Asiatic trip it will earn nearly twice as much even taking into account amortization, interest and capital charges.

The author saw the Still engine in England. It is an engine completely surrounded by auxiliaries. Someone has said that there is a new auxiliary born each day, and Still has them all. And with all these different auxiliaries there is only a comparatively small gain made. Theoretically it is good; practically it is altogether too complex. We want simplicity itself, if possible.

As to the efficiencies, it can be said that they approach more nearly the theoretical air cycle efficiency than any engines that have ever been built. And there is no reason why it should be

otherwise. We might suffer tremendously if the mechanical efficiencies were low. In Italy, a 7 by 11 Diesel engine, which is the size of the engine shown in the paper, without the low-pressure cylinder, running at 450 r.p.m. was found to give 13 hp. per cylinder. If this engine were to run at 400 r.p.m., the speed of the Sperry engine, it would give about 22 hp. in the two cylinders. In the Sperry engine, with these two cylinders and the low-pressure cylinder, we have 220 b.hp. It has been run a great many hours at 180 b.hp. This is about seven times as much power as the Diesel engine. But there is not seven times as much machinery operating, simply one more piston. Cards have been taken repeatedly from the Sperry engine to check up the mechanical losses and these are about 10 per cent. In larger engines, between 40 and 50 per cent thermal efficiency at the shaft is anticipated.

Now, as to large engines. What are the limitations on Diesel engines today? They are cylinder size and thickness of cylinder wall. It is impossible to go much above $1\frac{1}{4}$ in. of cylinder wall, nor much above 30 in. in diameter. With this in view and with 29-in. cylinders running 800 or 900 ft. piston speed, and leaving only four of these combustion cylinders in line, the layouts indicate that we would obtain an 8,000-hp. engine. With six combustion cylinders in line, this would be 12,000 hp. So there is no trouble about the size. Dr. Lucke, of Columbia, has been good enough to say that the compound has solved the problem of making large oil engines.

In reply to Mr. Shaw, the author wrote as follows:

"I wish to thank Mr. Shaw for his consideration of the paper. Mr. Shaw falls into the almost universal error in visualizing combustion engine compounding. He takes a normal Diesel card of the *William Penn* engines and extends the expansion down to one-half of the usual pressure at which the exhaust valve opens, hoping thereby to set forth a convincing argument against compounding. What he actually succeeds in doing is to produce a real case of what he characterizes as 'toe loss.' In applying this expression to Fig. 2, our critic seems to have overlooked such an insignificant point as the line *XY*. The difference between the two propositions is as great as the poles. Thanks to a little clearer vision, our expansion is not confined to the combustion cylinder. This is not the first, nor probably the hundredth time that I have been confronted with this erroneous explanation of what I am doing. Haven't I made myself clear in the paper that it is just this attempt to do low-pressure work in a high-pressure cylinder that is condemned and has

been the compelling reason for adoption of the compound principle in steam as well as combustion engines?

"I am in most hearty agreement with Mr. Shaw that were this all we did in compounding, it would be even more useless than he states it to be. This may be considered on a par with most of the other arguments set forth by the writer, and with no little elaboration of detail. Again, the argument is about on a par with the argument of Mr. Shaw's second paragraph, where he seems to think that the only gain in weight we have is the double use of the same low-pressure cylinder for two combustion cylinders 'provided the same general construction is adhered to,' etc. But isn't this begging the question? Is the same general construction adhered to? Hasn't our critic missed several other little points, such as the compound combustion cylinder being enabled to handle a great many times the weight of air compared with the simple Diesel — not only many times the weight of air, but many times the fuel with more complete combustion than in any simple engine? Again, the point of very great gain in chilled perimeter and surface per unit volume of the power gases. A number of other definite differences between the compound and simple engine I feel are quite completely set forth in the paper.

"Now of course, if our critic refuses to recognize practically all the paramount differences between the two engines and can only see 'back pressures' in the expansion cards, I do not believe it is worth while to follow much further the old and familiar extollation of heavy Diesels, although after some thirty years some recognition is evinced in his discussion of the desirability of starting to hack away in the effort to reduce these ponderous structures.

"The expression 'back pressures' appears at several points as being the great argument against compounding. Is it possible that Mr. Shaw sets this forth in the nature of a discovery, or as specially applicable to the present case, when, as a matter of fact, this is, and always has been, one of the basic considerations. A compound engine of any kind invariably develops back pressure on the piston of the next higher stage. All of these facts have had to be considered in compounding steam. In the early history of this earlier art all of these objections were doubtless dragged forth with many others, groomed and arrayed in opposition to the unthinkable change from simple to compound. Now, as a matter of fact, not only have none of these features been found prohibitive, but they have been so minimized as not even to be serious, and this has

been the proceeding in the present instance. Naturally there has not been a moment in the three decades in which this work has been pursued that all of these — back pressure, losses in ports, through valves, etc. — have not been prominently in mind and means not only sought, but found, to minimize them.

"With reference to the comparison made between Figs. 5 and 6, where the constant volume and constant pressure cycle are shown, doubt is expressed if the theoretical card is anywhere nearly approached in practice. We believe that Fig. 12 is an answer to this.

"The predictions as to shocks, strains and heat troubles are on a par with many other of the imaginary troubles. The facts are there is nothing that approaches shocks in the high intensity burning here shown. These imaginary troubles will always remain the reason of the timid for not taking a forward step. If our critic could be enticed to leave the beaten path and take an excursion into some very interesting research on this phase of the problem, he would in all probability be jarred by finding that the very time factor that so alarms him carries with it its own remedy in relation to its actual effect on the heat gradient and heat losses. Mr. Ricardo of England has evidently gotten sight of this phase of these interesting phenomena.

"There is no lack in the flexibility of the engine employing the high-intensity burning card.

"With reference to the comparison of the compound with the simple engine, inasmuch as there is no such thing as a previous compound combustion engine to compare with, I am perfectly justified in referring all of the mean effective pressures back to the only thing that previous engines do have, viz., the combustion piston area, and the evident value in securing ten times the net mean effective pressures to the crank for each fuel injection, together with a very much better distribution of the power application to the crank from any single injection, I feel needs no further emphasis here.

"In connection with the generating sets referred to in the *Sperryscope*, we cannot agree that the engine that is lighter than the standard General Electric or Westinghouse generator to which it is directly coupled is not exceptional, especially considering that this applies to the ordinary run of engines, designed with no thought whatever of lightness. Fortunately we do not have to compare the compound with other than trunk engines to bring into strong contrast the gain in weight. We all know of many instances of these engines in service running between 150 and 350 lb. to the

b.hp., many of them running at much higher piston speed than the comparatively low piston speed of 735 ft. employed in our 7 in. engine. However, when employing the same class of refinements as used in submarine engines, the weight comes down to a very low figure, as indicated in the paper.

"A point not emphasized in the paper is that in the compound much larger power units can be made in trunk engines than possible with simple engines, the controlling factor covering this point, as is well known, being the relation of combustion piston and the wrist pin bearing diameters. This gain follows from the fact that a given combustion cylinder in the compound accommodates from four to six times the fuel with a corresponding gain in power for the size. Naturally the alternative of this is that with the compound we do not have to resort to crossheads until we reach four to six times the power of the simple engine per combustion cylinder at the same piston speed and revolutions. It is felt that these are definite gains that cannot be neglected. The crosshead type of engine is just as applicable to compounds as to simple engines. The builders of crosshead engines are driven to adopt this from the limitations given above, whereas so long as excess slipper area is available with ample lubrication, and it shows practically no wear, many of our leading engineers are commencing to believe that the trunk engine is superior, and we know that it is lighter.

"With reference to the heavy engines, the writer makes it clear that a plurality of unit engines is necessary in a ship, but in all instances twin screws have to be resorted to, where a single screw would be much better all round. It is believed that plural units with their simple slow-speed gears will forge ahead and win the place that they deserve, especially in view of the complete safeguards of the air-gap clutch, which our critic seems to overlook, making the most completely maneuverable ship yet produced. With one of the engines of Fig. 24 reversed, the control of the ship rivals the reciprocating steam drive and has the added advantage of distant or bridge control. The fact that a few watts operate the clutches affords a complete solution to this problem. Shipping is already beginning to appreciate these advantages.

"As to the generator in Fig. 17, this being a photograph of a foreign-built engine and not one of Mr. Shaw's construction, there seems to be doubt as to the accuracy of the writer's remarks concerning it. Halfway measures are not usually adopted. A similar photograph of the Maume engines, which are still larger than the

one of Fig. 17, where a full-load generator was used to my knowledge, shows about the same contrast. But suppose the statement to be correct. Does Mr. Shaw think for a moment that if the generator shown were to be multiplied in size by the cube root of 2 the contrast between Figs. 16 and 17 would be substantially altered?

"If present signs are any indication, it will soon be possible for our critics to satisfy themselves as to all points of efficiency and economy of these engines under practical conditions of operation. This, it seems to me, should settle all controversial points and while the engines will probably not be above criticism, I believe they will serve to mark a definite turning point in combustion-engine construction. Meanwhile I agree with the discussion that it is to be regretted that I did not have the performance, say the fuel consumption, of a larger than 7-in. cylinder to base reports on when the paper was written. Larger engines are, however, usually better than smaller engines and certainly much better than minute engines in this regard. The smallness of the 7-in. cylinder has been referred to several times, but let us examine the performance of this little engine. It has given 0.302 and 0.305 lb. of 16 to 22 Baumé fuel per indicated horsepower-hour in spite of the fact that it has all the imperfections of an experimental engine. This we do not hesitate to submit as being fairly good performance for this size machine, being about the same as recently reported on the *William Penn* engines with their ponderous $29\frac{1}{8}$ in. cylinders. Entirely unnecessary wire drawing exists at two points in this engine, which was found to rise to larger proportions than was expected and has been eliminated in later designs. Other than this, the mechanical losses in this engine are extremely low, as referred to in the paper.

"From many valued sources outside of this discussion I have received words of commendation for the work. Among my critics there have also been, of course, some of the 'die-hard' class who profess to see no virtue whatever in compounding, everything about it being wrong. To these, is it possible to emphasize too strongly the facts above referred to, — the fuel consumption per indicator-diagram unit? If the thermal units are being excessively absorbed by the chilled walls or going astray in the many ways claimed by these critics, they most emphatically would never reach the indicator, this being, I feel, a more complete answer to the various points raised than is sometimes realized."

No. 1808

COMMERCIAL OPERATION OF AIRPLANES

By L. B. Lent, New York, N. Y.
Member of the Society

In order that those interested in the operation of airplanes for commercial use may have an idea of what may be expected of a properly organized and operated service, the author presents in this paper an analysis of the record of the Air Mail Service of the United States Post Office Department for the period October 1, 1920–September 15, 1921. The operating record, the cost of operation, and the life and maintenance of planes and of engines give a basis upon which the author points out possible improvements, the most important in present practice being the use of efficient commercial planes equipped with a thoroughly reliable power plant. In regard to cost of operation the author states that the total operating cost should not exceed seventy cents per plane-mile for single-engine planes of not over 400 hp.

IT IS the purpose of the author to present in this paper some information which may be found useful by those interested in the operation of airplanes as vehicles of transportation, presumably for profit.

2 Up to the present time there has been comparatively little such commercial service in this country, and consequently there have been very few data developed which might be useful in an investigation and analysis of this important subject.

3 Commercial flying in Europe is developing very fast, but the conditions under which it is performed make it difficult to apply the information gained to operations in this country. A large part of the foreign service is devoted to carrying passengers and the operations are in many cases partly supported by government subsidies, and nearly all of the elements of cost are such as to make it difficult to apply the results to any proposed operation in the United States.

4 Fortunately, however, there has been a service in operation in the United States since May 15, 1918, which has rapidly developed

Presented at the Annual Meeting, December 1921, of The American Society of Mechanical Engineers.

into what may be properly characterized as the largest commercial operation of airplanes in the world namely, the Air Mail Service operated by the Post Office Department of the United States Government.

5 This service is operated by a purely civilian personnel and is very similar in nearly all respects to that which would be effective in a purely commercial service. Moreover the records of this service have been carefully kept during its entire existence, and if properly interpreted can be used as a basis for estimating the probable performance and costs of any commercial service of the immediate future.

6 It is believed, therefore, that a better and more accurate idea of what may be expected of a properly organized and operated commercial service can best be gained by intelligent analysis of the record of the Air Mail Service. While many are no doubt familiar with this record, it is nevertheless presented herewith in as condensed form as possible in order that the reader may have in hand all of the information upon which the opinions and conclusions stated hereinafter are based.

7 While the Mail Service has been in operation since May 15, 1918, it has developed from an experimental service between New York and Washington, D. C., to the transcontinental service now in daily operation between New York and San Francisco. Without going into the history of this growth, suffice it to say that in the opinion of the author an analysis of operations of the last year, during which time the transcontinental service has been running, and particularly the last six months, will best represent present stabilized conditions.

8 The extension from Omaha to San Francisco was opened and the transcontinental service begun on Sept. 15, 1920. For purpose of discussion the period between Oct. 1, 1920, and Sept. 30, 1921, is chosen.

9 The record of the first few months of this service, during which time operating conditions were far from being ideal, was not as good as the latter part of the yearly period. This record, of course, was largely influenced by the fact that the first few months of operation were winter months, during which the service records of a well-established line are sure to suffer. However, the full record is given and in many cases attention is drawn to the record of the last six months as being perhaps a better indication of what might be expected of a well-established commercial line.

ORGANIZATION

10 The Transcontinental Air Mail Service operates daily in each direction between New York and San Francisco, an air-line distance of 2630 miles and a total daily mileage of 5260 miles. The various flying fields and the principal buildings located on each, are as follows:

New York (Hazelhurst)	1 Hangar
	1 Storage hangar
	1 Office building
Bellefonte, Pa.	1 Hangar
Cleveland, Ohio.	1 Hangar
	1 Workshop
	1 Office
Bryan, Ohio.	1 Hangar
Chicago, Ill.	1 Large repair hangar
	1 Storage hangar
	1 Operating hangar
	1 Office
	1 Stock house
	1 Test house with stands
	1 Oil house
Iowa City, Ia.	1 Office; stock room and repair building
	1 Tent hangar
Omaha, Neb.	1 Hangar
North Platte, Neb.	1 Hangar
Cheyenne, Wyo.	1 Hangar
Rock Springs, Wyo.	1 Tent hangar
Salt Lake City, Utah.	1 Hangar
Elko, Nev.	1 Tent hangar
Reno, Nev.	1 Hangar
San Francisco, Cal.	1 Hangar
	1 Office

11 At each of the major stations, where permanent hangars are located, there is a considerable amount of equipment consisting largely of spare parts for both planes and engines, workshops with the necessary tool equipment, trucks, and motorcycles, and in some cases caterpillar tractors, equipment for storing and handling gas and oil, and office equipment (in some cases in a separate building).

12 Communication between fields east of Chicago was originally maintained over leased telegraph wires, but during the past year this service has been discontinued and wireless communication is now used exclusively over the entire service, which method has proved very satisfactory.

13 The major repair shop for repairs on both planes and engines is located at Chicago, but minor repairs and engine overhaul are carried on to a limited extent at all of the major fields.

14 The airplane equipment now used in this service has been standardized and all machines are the De Haviland planes equipped with Liberty 12 engines turned over to the Mail Service by the War and Navy Departments. These machines are rebuilt to accommo-

TABLE 1 PERSONNEL OF THE U. S. AIR MAIL SERVICE

Field	Supts.	Asst. supts.	Managers	Chief mechs.	Mechanics	Riggers	Helpers	Chauffeurs	Clerks	Stock clerks	Laborers	Watch-men	Radio operators	Car-penters	Pilots	Total
Headquarters	2	2	.....	.....	1	.....	.....	.....	17	.....	.....	.....	9	.....	.....	28
Repair Depot	1	.....	.....	1	33	.....	32	3	4	3	8	2	.....	1	.....	87
Warehouse	1	.....	.....	2	4	.....	.....	4	5	6	9	6	.....	2	.....	39
College Park	.....	.....	1	.....	1	.....	.....	.....	.....	.....	.....	3	.....	.....	.....	5
St. Louis	.....	.....	1	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	1
Hazelhurst	1	.....	1	1	16	1	6	.....	2	1	.....	2	2	.....	4	37
Bellefonte	.....	.....	1	.....	3	1	3	.....	1	1	1	2	2	.....	.....	15
Cleveland	.....	.....	1	1	13	3	1	1	1	2	2	2	.....	.....	5	32
Cincinnati	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	2	.....	.....	2
Bryan	.....	.....	1	.....	1	.....	1	.....	.....	.....	.....	.....	1	.....	.....	4
Chicago	.....	1	.....	1	12	2	2	1	2	1	5	1	.....	.....	9	37
Iowa City	.....	.....	1	.....	3	.....	.....	.....	1	.....	.....	1	2	.....	1	9
Omaha	1	.....	1	1	14	1	2	.....	3	1	.....	2	2	.....	5	33
North Platte	.....	.....	1	.....	.....	1	4	.....	.....	.....	.....	2	2	.....	1	11
Cheyenne	.....	1	.....	.....	11	1	10	.....	1	1	.....	1	2	.....	5	33
Rawlins	.....	.....	1	.....	1	.....	.....	.....	1	.....	.....	.....	.....	.....	.....	3
Rock Springs	.....	.....	1	.....	4	.....	5	.....	.....	1	.....	2	2	.....	2	17
Salt Lake	.....	1	.....	1	12	.....	2	2	1	1	.....	1	2	.....	5	28
Elko	.....	.....	1	1	2	.....	4	.....	1	.....	.....	2	1	.....	1	13
Reno	.....	.....	1	.....	9	1	6	.....	1	.....	.....	1	1	.....	3	23
Frisco	1	.....	1	1	6	2	3	.....	3	1	.....	2	.....	1	3	24
Total	7	5	14	10	146	13	81	11	44	19	25	32	27	4	44	482

date the mail load in the cockpit in front of the pilot, and strengthened to meet the hard service required. A considerable amount of this equipment was turned over as indicated, and is stored in two buildings in Newark, N. J. This constitutes the major source of supplies for the entire service.

15 The total investment of the Air Mail Service is about $800,000, of which $133,000 is for buildings, trucks, tools, etc., the remainder being for airplanes and engines.

PERSONNEL

16 The personnel employed is shown in Table 1. In this table two radio operators are shown in Cincinnati, Ohio. There is no field there, but these two men assist in this work and are paid by the Air Mail Service. All radio operators handle a considerable amount of work for the Post Office Department other than purely Air Mail communication, but are paid out of Air Mail funds.

EQUIPMENT

17 On Oct. 1, 1920, the service was equipped with about the same total number of planes as at the end of the year, planes damaged beyond repair being replaced by others taken from warehouse stock and rebuilt for the service. It is obvious that the actual number in serviceable condition and under repair may vary from day to day, but the following are fair average figures for the year: In serviceable condition, 50; under repair, 18; awaiting repair, 30. During the year, 27 planes were damaged beyond repair and salvaged, while 48 new planes were placed in service and about 20 experimental types retired. All planes now in use are the rebuilt De Haviland-4 type.

18 Engine equipment has always been in excess of actual need, because of the large number of Liberty engines turned over by the Army. It is fair to assume that the service could be as well operated with a considerably less number of engines as hereinafter shown. The total number of Liberty-12A (Army high-compression) and Liberty-12N (Navy low-compression), developing about 400 and 350 hp., respectively, in use is about as follows: Engines in serviceable condition, in planes and as spares, 150; engines under and awaiting repair, 350. The foregoing figures do not represent the engine requirements for such a service.

TABLE 2 OPERATING RECORD OF THE U. S. AIR MAIL SERVICE. OCT., 1920—SEPT., 1921

Month	Trips Possible (Scheduled)	Trips Attempted	Trips Defaulted	Trips Uncompleted	Weather Encountered		Mileage Possible	Miles Traveled	Per Cent of Performance[1]	Mail Carried, Lb.	Forced Landings Due to	
					Trips in fog	Trips clear					Mech.	Other
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)	(13)
1920												
October	750	593	157	57	190	143	154,700	123,274	79.68	89,541[4]	54	59
November	758	575	183	77	230	345	156,076	114,750	73.50	87,302	85	69
December	884	662	222	105	347	315	178,776	127,306	71.21	89,942	89	138
January	850	697	153	87	253	444	171,900	132,679	77.18	84,435	117	131
February	782	660	122	71	310	350	159,238	130,431	81.90	88,135	82	122
March	918	871	47	59	351	520	185,652	171,593	92.42	110,117	64	123
April	884	837	47	36	370	467	178,776	171,156	95.73	117,778	79	107
May[2]	850	833	17	20	320	513	171,900	168,397	97.96	115,073	72	81
June[3]	832	829	3	10	268	561	168,636	166,956	99.00	105,838	57	51
July	624	623	1	6	65	558	131,450	130,555	99.31	77,276	34	29
August	693	689	4	13	129	560	136,974	134,549	98.22	84,680	32	30
September	657	651	6	8	180	471	127,706	125,914	98.59	88,401	13	26
Total	9482	8520	962	549	3013	5247	1,921,784	1,697,560	88.33	1,138,518	778	966

[1] For last 6 months, 98 per cent.
[2] May 31 — Last day's service on New York–Washington Route.
[3] June 30 — Last day's service on St. Louis–Twin Cities Division.

19 An accurate estimate of the total engine requirements for this service can be based on the following: 22 engines are required for the 22 planes in daily service; at least 1 spare engine should be at each major station, making a total of 14; another supply of approximately 30 engines is necessary for those constantly in transit between the various fields and the overhaul stations, including the period of overhaul, making a total of 66 engines. If this service is properly organized a total of 75 engines is ample for successful operation of such a service.

OPERATING RECORD

20 The operating record for the past year is shown in Table 2. For a thorough understanding of these figures it is thought well to outline briefly the conditions under which the service was rendered.

21 The transcontinental course is divided into 13 legs of approximately 200 miles each with stations as given above. The route is divided into three major divisions — the eastern, extending from New York to Chicago; the central, from Chicago to Rock Springs, Wyo., and the western, from Rock Springs to San Francisco, Cal. Each division is in charge of a division superintendent, who has an assistant division superintendent. At each major field is a field manager, except on such fields as the division superintendent or assistant division superintendent may have his headquarters, in which case either one of these officials may act as field manager. The chief mechanic at each field is in charge of mechanical crews for both maintenance and repair. The principal repair shop at Chicago is in charge of a superintendent of repairs, and the warehouses at Newark are likewise in charge of a warehouse superintendent.

22 It has been found that in the western trip the flight is hampered by prevailing westerly head winds, whereas the eastern trip is of course assisted by such conditions. Inasmuch as the supply of gas for each ship is approximately only four hours at cruising speed, it becomes necessary in many flights to stop at intermediate stations for service. Many times coming east, planes fly a major leg of approximately 400 miles. The number of such trips per month is somewhat varied, depending upon conditions.

23 While the available space in each plane accommodates 400 lb. of mail, the average amount carried per trip for the entire year, was less than 150 lb. per plane per trip. These planes carry

a pilot only, who flies a leg in one direction each day, returning to his home station the next day and laying off the third day. This schedule is of course subject to change, which occasionally results in a pilot doing more mileage than indicated.

24 The total flying time per day varies with the weather conditions, but the average speed for the entire year was 86.3 miles per hour; for the last 6 months (during better weather conditions) the average speed was 87.88 miles per hour. Up to the present time all flights have been made during the day. An indication of what is possible in a continuous trip is the record established on Feb. 22 and 23, when the mail was carried from San Francisco to New York in elapsed time of 33 hr. 20 min., or a total flying time of 26 hr. 50 min. The fastest scheduled train time between these two points is 92 hr. The eastbound trip from Chicago to New York has been made in 5 hr. 30 min. The flight from Salt Lake City to San Francisco was made on Oct. 14, 1921, in 6 hr. 1 min. elapsed time, or a total flying time of 5 hr. 32 min. over a distance of 624 miles. The fastest train between these two points is scheduled for 24 hr. 15 min. It follows that, for the transcontinental distance, daylight flying only will cut train time about in half, and with night flying the time is cut to about one-third.

25 The operation record discloses one feature of special merit—given under "Percentage of Performance." It will be noted that the percentage of performance for the year was 88.33 per cent. For the last 6 months this average was 98 per cent, of which 96 per cent was actually completed on scheduled time. In view of the fact that the Pennsylvania Railroad in its printed timetables boasts of a train performance of 95.6 per cent on time, the foregoing record for the last 6 months is truly remarkable. It will of course be suspected that performance is largely affected by weather conditions, as it is clearly shown in the table of performance, but it should also be borne in mind that during the first few months the transcontinental mail service did not have proper or complete equipment, and that almost the entire personnel west of Omaha was new to the service.

26 The effect of weather on the service is illustrated in Fig. 1, which shows the record from the start of the service in May, 1918, to the end of September, 1921. It will be noted that defaulted trips and the number of forced landings show a decided increase during the winter months. An interesting fact is that the number of forced landings from mechanical trouble also shows an increase, which

would indicate the effect of cold weather on the quality of work done by mechanics.

27 During the year ten pilots and one mechanic were killed — four of the former during the last six months. This is equivalent to one pilot killed for each 169,756 miles flown.

28 During the year 27 planes were so badly wrecked that they were salvaged instead of repaired, and there were 145 forced landings which resulted in damage to planes necessitating repair work; of these about 100 required major repairs. Obviously some were damaged more than once during the year. A further analysis

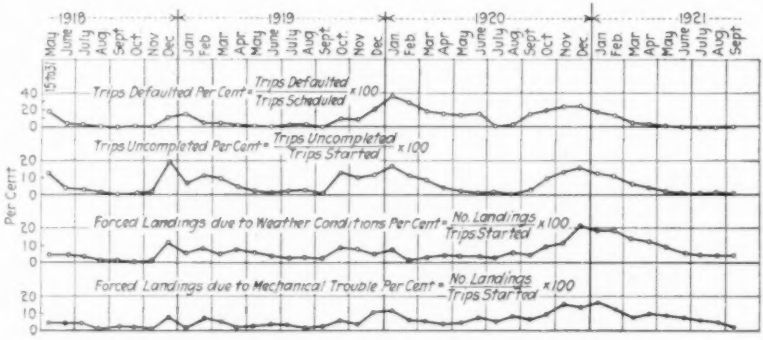


FIG. 1 SEASONAL EFFECT ON PERFORMANCE OF AIRPLANES OF U. S. AIR MAIL SERVICE

of this as affecting the cost of operation is given in a succeeding paragraph.

29 The large number of actual forced landings from mechanical trouble and other causes will no doubt attract attention as being excessive, and it should be noted that of the total number of forced landings only 172 resulted in damage to the planes. This is one plane damaged for every 10 forced landings.

30 It should, in fairness, be pointed out that all landings other than those regularly scheduled, even though made on an intermediate Air Mail field, are counted as forced landings.

31 A pilot flying against a heavy wind may "set-down" on an intermediate field for gasoline rather than take a chance of his supply lasting out the scheduled trip. It has also been necessary to fly away from New York on snow skids, stopping at Bustleton, Pa., to change skids to wheels so as to land on bare ground at Washington, D. C.

32 An enumerator of forced landings on a mileage basis is shown in Table 3. In interpreting these figures, the several kinds of forced landings mentioned above should be borne in mind. The effect of damage to planes on the cost of the service is discussed later.

TABLE 3 MILES PER FORCED LANDING AND PER "CRASH"

Period	Mechanical trouble	Other causes	All causes	Miles per crash
1920				
October	2285	2900	1090	6180
November	1350	1665	745	7760
December	1430	920	560	5040
1921				
January	1130	1015	535	5180
February	1590	1070	640	8975
March	2680	1390	918	10360
April	2165	1595	920	17000
May	2340	2080	1105	15309
June	2930	3270	1545	41700
July	3830	4500	2070	14506
August	4210	4480	2170	67275
September	9680	4840	3235	31478
Average for year	2160	1740	963	17560
Average for last six months				31211

COST OF OPERATION

33 The consolidated costs of operating this service for the yearly period chosen are shown in Table 4. For an intelligent understanding of this table, the headings are defined as follows:

1 *Gasoline Cost.* Total cost of gasoline in planes as fuel for motors.

2 *Grease and Oil.* Total cost of motor oil and grease placed in the plane for operating.

3 *Repairs and Accessories.* Includes the labor and material costs of repair of planes and motors and their auxiliaries. Material costs are based on original wartime prices charged the Air Service and the prevalent market prices of material now purchased.

4 *Miscellaneous.* Includes travel expenses of all employees, stationery, small-tool equipment, the policing of grounds and some permanent improvements.

5 *Motorcycles and Trucks.* The total cost of the care, operation and repair in labor and material of the entire motor transportation equipment.

6 *Rent, Light, Fuel, Telephone and Water.* This indicates the total cost of these items for each station, warehouse and building.

7 *Office Force and Watchmen.* The entire cost of all field-office personnel, stock clerks and watchmen, is included in this item. Supervisory personnel is considered as a part of the office force.

TABLE 4 CONSOLIDATED COSTS OF OPERATING U. S. AIR MAIL SERVICE FOR YEAR ENDING OCT. 1, 1921

Month	Gasoline	Grease and oil	Repairs and accessories	Miscellaneous	Motorcycles and trucks	Rent, light, fuel, power, telephone and water	Office force and watchmen	Warehouse	Pilots	Mechanics and helpers	Radio	Departmental overhead charge	Total
October, 1920...	$20,959.94	$3,840.85	$22,571.32	$16,264.81	$5,827.95	$1,813.91	$8,501.34	.....	$10,989.85	$21,204.47	.....	$5,448.06	$117,422.50
November	17,635.97	3,920.24	18,626.50	16,511.32	5,197.57	2,511.17	9,081.01	.....	16,460.01	19,510.24	.....	6,289.69	115,743.72
December	17,783.40	3,814.04	21,072.46	21,529.14	5,199.62	2,954.73	12,310.29	.....	15,816.07	20,564.00	.....	4,031.01	125,094.76
January, 1921...	18,558.44	4,303.29	28,006.76	18,790.86	6,010.07	3,405.32	12,136.21	.....	15,432.13	19,583.76	.....	3,880.59	130,107.43
February	18,823.70	3,824.18	17,689.41	20,464.60	6,581.91	3,904.82	12,025.41	.....	16,246.74	18,388.37	4,744.81	3,775.30	126,469.25
March.....	22,896.17	4,970.33	33,531.52	17,827.86	6,832.30	3,792.54	11,270.34	.....	17,851.24	19,912.92	4,583.10	3,725.75	147,194.07
April.....	22,956.61	5,554.75	31,321.32	12,829.73	6,457.85	2,559.12	12,254.29	4,277.72	17,965.49	18,577.71	9,834.41	3,301.64	147,890.64
July	21,752.41	4,870.72	15,217.83	10,551.05	5,666.64	2,277.41	11,868.01	4,484.67	17,708.11	16,595.44	11,184.21	3,578.24	125,754.74
June.....	19,624.08	4,544.75	27,921.81	8,236.00	3,918.51	1,110.32	9,493.98	4,524.12	18,897.17	17,343.52	8,602.67	3,262.90	127,479.83
July.....	15,314.14	3,377.71	20,312.72	9,759.07	3,966.08	1,209.05	9,304.52	5,357.08	14,636.14	13,912.18	9,629.07	3,021.35	109,799.11
August.....	14,711.17	3,513.35	20,948.14	9,014.39	4,215.42	1,205.92	8,323.45	5,574.94	14,764.20	15,008.99	5,571.22	2,882.58	106,986.98
September.....	13,530.82	3,232.78	18,742.45	8,690.28	3,941.77	1,725.59	9,280.02	5,415.87	14,329.58	14,773.75	5,772.40	2,799 83	102,998.45
Totals.....	$224,546.85	$49,766.99	$275,962.24	$170,469.11	$63,815.69	$28,469.90	$125,848.87	$29,634.40	$191,096.73	$215,375.35	$59,921.89	$45,996.94	$1,482,921.48
Per Cent of Total Cost	15.0	3.35	18.6	11.5	4.3	1.91	8.5	2.0	12.8	14.5	4.04	3.1	

8 *Warehouse.* The total cost of maintaining and operating the main warehouses at Newark is represented by this item.

9 *Pilots.* The cost of retaining pilots at a base salary of $2000 per annum, plus their flown mileage at 5, 6 or 7 cents per mile, varying with the country they fly over.

10 *Helpers and Mechanics.* This item represents the total cost of all mechanics and helpers and common labor employed in the operation and maintenance of planes not in the repair shops, together with personnel necessary for the care and maintenance of buildings and field areas.

11 *Radio.* The total cost of the operation and maintenance of the entire radio system is represented by this figure.

12 *Departmental Overhead Charges.* Consisting of the entire supervisory personnel at Headquarters in Washington, with the necessary clerical personnel for the record keeping and other office work. 20 per cent of the Second Assistant Postmaster General's time is charged.

34 In further explanation it should be pointed out that the cost of operating the Newark warehouse previous to April, 1921, (when it was charged as a separate item) was distributed in the miscellaneous account (column 4). The same thing is true of the cost of operating the radio department.

35 It will be noted that there is no charge for interest on investment. This is of course a proper charge in any commercial service and should be added in any estimated costs. Six per cent interest on investment on planes and engines at their wartime cost was charged in the Mail Service and an attempt made to charge interest against investment in buildings, motorcycles, trucks and other equipment; but since a large amount of this equipment was turned over by the War Department and its value in a large degree unknown, it was thought that any interest charge was more of an assumption than an actuality. By making a proper interest charge against investment and adding this to the total costs shown in the table, it is thought that the figures resulting are safe and conservative for estimating the operating costs for any similar commercial line. In fact, the figures shown can be very much bettered if advantage is taken of the experience gained in the Mail Service.

36 An analysis and consolidation of the figures given in this table are shown in Table 5, which give the service and unit costs and the costs per mile for the year. In this table the total costs are divided into three general headings, namely, Overhead, Flying and Maintenance. Under the head of Overhead is included departmental overhead, office force and watchmen, motorcycles and trucks, rent, light, fuel, power, telephone, water and radio.

37 Maintenance consists of miscellaneous, mechanics, helpers,

repairs and accessories, and warehouse charges. Flying consists of gasoline, grease and oil, and pilots.

38 It will be noted that the mileage as shown in column 9, Table 4 and in column 3, Table 5, do not agree. The former figure is miles traveled in regular mail trips and the latter is "total miles" flown, which includes ferry trips, test flights and retrieving planes. The "total miles" is the proper basis for estimating costs per mile.

TABLE 5 ANALYSIS AND CONSOLIDATION OF FIGURES GIVEN IN TABLE 4 FOR PERIOD OCT. 1, 1920–SEPT. 30, 1921

	Service and Unit Cost					Cost Per Mile, Cents		
	Gasoline gal. (1)	Total Time, hr.-min. (2)	Total miles (3)	Cost per hour (4)	Cost per mile (5)	Overhead (6)	Flying (7)	Maintenance (8)
1920								
October	60,010	1,852-56	154,486	$66.71	$0.80	$0.18	$0.23	$0.39
November	50,573	1,651-20	139,759	73.57	0.87	0.21	0.27	0.39
December	49,844	1,569-28	131,040	83.60	1.00	0.23	0.29	0.48
1921								
January	51,427	1,668-51	139,609	81.78	0.98	0.23	0.27	0.48
February	68,698	1,733-04	153,655	76.08	0.92	0.25	0.27	0.40
March	66,929	2,103-36	186,625	73.18	0.82	0.19	0.25	0.38
April	66,854	2,175-04	186,950	67.98	0.79	0.18	0.25	0.36
May	64,322	2,041-29	181,216	61.08	0.69	0.19	0.24	0.26
June	57,873	2,148-27	185,091	59.33	0.69	0.15	0.23	0.31
July	48,625	1,644-26	148,684	66.76	0.73	0.18	0.22	0.33
August	47,818	1,696-30	149,432	63.06	0.71	0.16	0.22	0.33
September	43,953	1,667-07	148,099	61.72	0.70	0.17	0.21	0.32
Total	676,926	21,952-18	1,894,646	$69.57 Avge.	$0.808 for year	$0 1933	$0.246	$0.368

Total average for year.. 80.66 cents per mile
Avge. for last 6 months....$0.1716 $0.2282 $0.3182
Total Average for last 6 months 71.83 cents per mile

Average speed for the year $= \frac{\text{Col. 3}}{\text{Col. 2}} = 86.3$ miles per hour

Average speed for the last 6 months = 87.88 miles per hour

39 In further explanation of these tables it should be stated that the New York-Washington division was discontinued on May 31, 1921. The St. Louis-Twin Cities division was discontinued June 30, 1921.

40 An examination of Table 4 will show that the largest items of cost are gasoline (15.0 per cent of the total), repairs and accessories (18.6 per cent), pilots (12.8 per cent) and mechanics and helpers (14.5 per cent). Considering these separately, it is seen that the

gasoline cost cannot be materially reduced by the use of more economical engines, for even if engine economy is increased, say, 10 per cent (which is difficult), the total reduction would only be one-tenth of 15 per cent, or 1.5 per cent of the total.

41 The largest item, repairs and accessories (column 3), is susceptible of considerable reduction. The elimination of forced landings and the damage resulting therefrom would largely reduce this charge. Fair wear and tear on places is relatively very small.

42 The repair and maintenance of engines would then constitute a large part of this item, and this charge would be largely reduced by the elimination of forced landings as discussed hereinafter.

43 The basis on which pilots are paid, namely, $2000 per annum, plus 5, 6, and 7 cents per mile flown, has proved to be very satisfactory — more so than any other basis tried by the Mail Service. In all cases it supplies the men with a living wage and develops efficient service, which is rewarded by increased pay.

44 The pay for chief mechanics in the Mail Service averages $2000 per annum; other mechanics, from $1400 to $1800 per annum; helpers, from $1100 to $1400 per annum; and watchman and helpers, from $900 to $1200 per annum.

45 An explanation of column 6 is essential. In most of the major fields the Mail Service was supplied with flying-field, and in some cases with hangar, facilities, by the cities in which they are located, without charge therefor. Rent for field and hangar facilities is paid at New York, Bellefonte, Pa., Cleveland, Ohio, Bryan, Ohio, Chicago, Ill., and College Park, Md. (training station). At all other cities west of Chicago no rent is paid.

46 The item of cost is therefore comparatively low and should be modified if used for estimating similar costs of a commercial line, unless similar arrangements could be made with various municipalities for use of municipal airdromes. If commercial companies have to purchase flying fields as well as field equipment, it would of course add largely to the capital costs, as well as the operating charges. It is believed, however, that very favorable arrangements can be made with most of the large cities in the United States for proper landing-field facilities.

47 The analysis shows that the average yearly operating cost per flying mile was 80.66 cents and for the last six months, 71.83 cents.

48 Regardless of the effect of severe winter weather on the cost of flying, it may be noted that the cost is steadily decreasing month by month, and it is confidently expected in the Mail Service, and is the firm conviction of the author, that the cost shown can be considerably reduced by the application of lessons learned in this Service.

49 As mentioned herein, if the cost is reduced to a ton-mile basis, the figures resulting will be unreliable; a fair basis of comparison is on a plane-mile basis, for it costs but little more to fly a plane loaded to its capacity than it does to fly it empty. It is a safe statement that under the conditions prevailing in the Mail Service, the cost per plane-mile will soon be reduced to the neighborhood of 60 cents.

50 With proper equipment and organization, a commercial service, using single-engine planes of not over 400 hp. each and flying them only one round trip per day over a distance equivalent to that of the transcontinental service, can be accomplished for a total cost not exceeding 50 cents per flying mile. This is especially true if more than one round trip per day is flown.

LIFE AND MAINTENANCE OF PLANES

51 It may be noticed that in the foregoing no account is made of depreciation. In the present service no plane has remained intact long enough to wear out and so determine even approximately a rate of depreciation. Before a plane wears out most or all of the parts are replaced and charged up under that heading, or else it is so completely destroyed as to be retired from service, charged off entirely, with credit for such material as can be salvaged for use in repairing other planes. A fairly accurate idea of the rate of this destruction may be had from the following data:

52 During the year 27 planes were so badly damaged that repairs were not undertaken, these planes being salvaged. Of the many forced landings, 145 resulted in "crashes" necessitating repairs, of which about 100 necessitated major repair work. The number of "crashes" is thus about 10 per cent of the total number of forced landings. The average for the year is one "crash" for each 17,560 miles flown with mail; and for the last six months, 31,211 miles.

53 The number of planes required as original equipment to operate such a service as the Mail Service and the rate and cost of replacement may be estimated as follows:

1 plane for each of the 22 daily flights	22
1 spare plane for each daily flight, to be used in case of "last-minute" trouble with scheduled plane	22
1 spare plane at each of the 14 fields which may be in transit to and from repair shops and under repair	14
Total	58

54 Mail Service experience shows that after a plane has had about 450 hours in the air it is necessary to lay it up for a thorough inspection and reconditioning. This operation involves some labor and material (usually new wing covers, etc.) and costs an average of $600 per plane. The total time required for such an operation at present is about two months. Since the flying hours per year is 21,952, the total number of such overhauls is 21,952 ÷ 450 or 49. The record also shows that about 100 planes were so damaged as to require repairs. The average cost of these repairs was about $1000 each and the average time about two months per plane. The present rate of repair and overhaul for the entire Mail Service is about 11 to 12 planes per month.

55 It is thus seen that about 14 to 15 planes (about one for each 14 fields) must be constantly traveling to and from the repair shops and undergoing repairs. It should be stated that some plane repairs are made at many fields as well as at the main repair shop in Chicago.

56 An analysis of the above shows that:

a	The initial plane equipment is about one (1) plane for each 82 miles of scheduled flight	
b	One (1) plane repair for each 16,975 scheduled miles flown	
c	One (1) plane reconditioned for each 34,000 scheduled miles flown	
d	One (1) complete "washout" for each 62,000 scheduled miles flown	
e	The cost of conditioning 49 planes at $600 each	= 1.8 cents per mile.
f	The cost of repairing 100 planes at $1000 each	= 6.0 cents per mile.
g	The cost of "washouts" for 27 planes at $4000 each	= 6.5 cents per mile.
h	Total cost for repairs and replacements	= 14.3 cents per mile.

57 This amounts to a total estimated cost of $238,000. Deducting this amount from the total in column 3 ($275,962), leaves $37,962, which it is fair to assume is the cost of engine overhauls and repairs, or about 2.25 cents per mile.

LIFE AND MAINTENANCE OF ENGINES

58 The record of engines actually replaced, repaired, salvaged, etc., is so involved in the records that it is difficult to discover the exact data. However, methods of maintanance, inspection and overhaul have been so developed and improved that it is more accurate to base an estimate on the present rate than to use total figures for the year.

59 Present methods of conditioning and caring for engines have resulted in an almost uniform service of 100 hours of running between each overhaul, unless an engine is damaged in some "crash." Since a total daily mileage with mail is about 5300 miles and the average speed is 86.3 miles per hour, the daily flying time is about 62 hours per day. To this must be added the time consumed in warming up engines (6 hours) and the time of engines on various test stands, so that the total daily time for the service is 80 engine-hours. This is equivalent to 0.8 engine replaced per day. Crashes, mechanical troubles and causes other than plain wear bring up the rate to just about 1.0 engine per flying day. While this may seem high, considering that only twenty-two planes fly regularly each day, it is not, for 100 flying hours per engine between overhauls is a very high figure and can only be attained by the most efficient and systematic conditioning and repair methods in force in the Mail Service. While about 300 engines were used, the whole 300 were not actually overhauled, as many new engines were placed in service during this period. The yearly demand, however, is at about a rate of 300 engines per year.

60 This running time of 100 hours between overhauls is true even after the second and sometimes the third overhaul, the necessary replacements being made at each overhaul.

61 The average cost of an engine overhaul is as follows:

One mechanic and one helper, 10 days, 8 hours each	80 hours
Washing and valve grinding by other men	8 hours
Carburetor work, cleaning and adjusting	4 hours
Ignition work	4 hours
Block testing	6 hours
Total	102 hours

62 This total time may vary 50 per cent each way, but it is a fair average. The average labor cost is about $75 per engine.

The average cost of the necessary material replacements, based on Government war prices, is $150. The average total cost is therefore $225 per engine. This cost may seem excessive, but is justified by the amount of careful and accurate work done, which is the real reason for 100 hours of further satisfactory service.

63 Right here may be a proper place to point out the economy resulting from such methods. More frequent overhauls and a less satisfactory service record are a sure sequel to a lack of attention to the power plant, which is *the* important element. To illustrate, consider briefly the procedure employed in the Air Mail Service. Even a new engine is not assumed to be in proper condition. Nothing is "taken for granted." The method of conditioning of such an engine, and in general all engines after repair, may be briefly described as follows:

64 Even new motors are at least partly torn down to inspect the condition and fit of cylinders, pistons, piston rings, crankshaft and pin bearings, gears, etc. Crankshafts are especially checked for alignment, end play and proper condition of thrust bearings. Carburetors are carefully inspected and calibrated for both mechanical perfection and proper functioning, as well as are generators, distributors, and other parts. The usual procedure is to put on every engine a set of carburetors, distributors, etc., which have already been inspected and calibrated in workrooms devoted to this special work. Such similar parts removed are turned into these departments for conditioning. All gaskets, especially those at carburetor and intake-manifold connections, are replaced with those whose condition is known to be good, thus preventing possible air leaks and improper carburation. Rubber-hose connections on gas, oil and water pipes are all replaced when necessary and it is usually necessary.

65 By using a timing disk, the correct timing of camshafts, valves and distributors is checked and the distributors for each bank of cylinders are synchronized both mechanically and electrically to within a quarter of a degree.

66 After a complete inspection and conditioning, the engine is run for about two hours at speeds varying from 300 to 1400 r.p.m. and also with wide-open throttle. After test, the engine is washed with kerosene and again inspected externally, and any faults which may have developed are corrected. All nuts are tightened, if necessary, all cotter pins are placed and securely fastened and the general condition made up to a *standard* and passed by the chief mechanic.

67 This more or less detailed description is given to emphasize the importance of a most thorough, methodical, scientific and accurate inspection and conditioning of airplane engines. The resulting service fully justifies the means.

68 The foregoing constitutes a presentation of the mail record for the year, together with its analysis and certain deductions from this record, based on the author's experience in, and knowledge of, the Air Mail Service. It is assumed that some of the lessons learned in this service, and perhaps some of the opinions as to what improvements are possible, will be interesting, and they are accordingly set forth in the following paragraphs.

IMPROVEMENTS POSSIBLE

69 First and foremost, it must be apparent that the actual pay load (or mail load) carried is very small for planes powered with 400 hp., and if the cost of operation be reduced to a ton-mile basis, the results are not encouraging. However, it must also be evident that the costs really apply to a specific number of plane-miles, and not ton-miles, for it costs almost exactly as much to fly a plane empty as it does loaded. The mail planes carried an average load throughout the year of only about 133 lb. per trip attempted. As previously mentioned, the maximum mail load possible to put into the space available in the fuselage of these converted war machines was about 400 lb. These machines are now being remodeled to carry 850 lb. of mail. The important point, however, is that in a plane designed for commercial service, using the same Liberty-12, 400-hp. engine, and *costing no more to operate*, the *pay load should be all of 2000 lb.*

70 Another important point to bear in mind is that the mail planes make *one* trip each way per day. A commercial service operating several trips per day could do so without greatly increased overhead costs and at a considerably reduced unit cost of service. Just what the reduction would be, would depend, of course, on the number of trips made. Hence it may be fairly said that a most apparent improvement is a use of planes designed for commercial service, those which will carry a maximum load economically over a reasonable distance at high speed.

71 The cruising speed of such a plane should not be much, if any, less than 100 miles per hour. Head winds of 30 to 40 miles per hour are sometimes encountered, which slow down the actual

ground speed by exactly that much, which means that the 100-miles-per-hour plane is traveling at (100–40) 60 miles, whereas a plane having an air speed of 130 miles per hour, or 30 per cent more, is doing (130–40) 90 miles per hour over the ground, or 50 per cent more than the 100-mile plane. Some carrying capacity should be sacrificed to speed in order to enable a plane to make time under adverse weather conditions. High speed also decreases the time required to fly a given distance and also decreases the total gas consumption, an important consideration, as will be seen.

72 Inasmuch as the fuel load is one of the factors determining the size and weight of the plane, it is important that the fuel be used as efficiently as possible. The actual cost of the fuel burned is not so important as the reduction in the total fuel weight for a given flight, for this saving in weight may be put into added cargo; or, more important, it can be put into added power-plant weight, which will, in large measure, prevent forced landings. Overall efficiency involves (*a*) reducing plane resistance, (*b*) increasing the loading per unit of wing area, (*c*) using a propeller of the highest possible efficiency, (*d*) running the engine, or engines, at nearly full power, and (*e*) using engines of high specific economy. The effect of a proper combination of the above elements is to reduce first cost by using relatively small machines to do the work of large ones, which in turn decreases capital costs and further reduces the operating expense.

73 The question of reducing plane resistance, that is, increasing the ratio of the lift to drag (L/D), seems to have been given serious attention only in military or racing machines and has been ignored on commercial machines, especially of the multi-engined type.

74 As the gross load which can be carried is equal to the thrust of the propeller multiplied by the ratio of the lift to drag (L/D) at a given speed, and as a low-resistance plane does not need to weigh more than one of higher resistance, it follows that the "usefulness" will be increased by increasing the lift-drag ratio, but in much greater proportion. For example, in an airplane making 100 miles per hour with a propeller efficiency of 80 per cent, the net available thrust is 3 lb. per engine hp. If the L/D ratio of such a plane is 6 at this speed, the total weight-carrying ability is 18 lb. per hp. If the weight empty is 10 lb. per hp., the net weight of the useful load is 8 lb. per hp. By increasing the lift-drag ratio to 9 instead of 6, the total weight-carrying ability will be 27 lb. per hp., and the net useful load 17 lb. per hp. instead of 8 lb. In other words, an increase of

50 per cent in the L/D ratio increases the useful load capacity 112 per cent, a gain by no means negligible and, in fact, not difficult to obtain.

75 Typical airplanes in use at the present are operating under conditions worse than instanced above. May we not, therefore, look for immediate improvement in this respect at least?

76 Real airplane-engine efficiency is not in fuel economy alone, although a low specific fuel consumption is essential. Contrary to more or less popular ideas, reliability is not a direct function of weight per horsepower only, for the mere addition of weight does not necessarily add reliability. As a matter of fact, the designers of recognized ability agree that airplane engines should be as light as possible. Lightness is an inherent quality of airplane-engine design and does not necessarily mean decreasing the factor of safety of important members below a safe figure. Making valves (for instance) much heavier would seriously interfere with their proper functioning as valves.

77 The best engines have a combined fuel and oil consumption of seldom less than 0.5 lb. per hp-hr. at full load, so that fuel and oil for a 5-hour flight is 2.5 lb. per hp., or more than the weight of a modern engine. It is therefore important that the fuel be consumed efficiently.

78 If the desire to "take off" quickly and climb rapidly is met, there must be much more power available than is required for economical cruising, and since all present types of engines have a higher *specific* fuel consumption when throttled, it follows that best results are to be obtained by some method which permits engine speed reduction with a nearly wide-open throttle. The variable-pitch propeller for single-engine machines is one solution of the problem.

79 Another solution is the multiple-unit power plant; that is, two or more power units driving a single propeller.

80 This brings us to a most important consideration in commercial operation, viz., the prevention of forced landings. The figures given in Table 3 are sufficiently striking to need little further comment, except to emphasize the fact that the elimination of forced landings will practically insure 100 per cent service all of the time and materially reduce the amount of the largest single item of cost, viz., that of repairs of planes and engines. Until we can fly planes from one airdrome to another with certainty, even at reduced speed, we cannot obtain the best service records nor the lowest operating costs.

81 The author ventures to predict that the next most successful departure from a single-engine machine will be the multiple-unit plant using a single propeller. And further, that such a power plant will not consist of separate complete engines geared to one "stick," but perhaps of two, three, or even more, banks of cylinders on a common crankcase, which will also house the necessary shafts, gears and clutches in this rigid structure. All the units of such a power plant could be used in "taking off" and climbing to a safe altitude, after which one or more could be shut down and held in reserve to be put into service if required. Engine and propeller speeds in such a unit could be made such as to give maximum combined efficiency under normal operating conditions. The weight of fuel saved in such an efficient combination, over a reasonable length of flight, will compensate for the added weight of the "relay" units of such a power plant.

82 The experience of the Mail Service, as well as that of others, is that a landing chassis of ample strength saves many a "crash" in a forced landing, as well as in a hard landing on the flying field. If the landing gear stays together, much damage is averted which otherwise might be serious.

83 Large wheels and large tires have proved to be most useful, especially in rough and soft ground. In fact, the Mail Service uses Handley-Page bomber wheels on the DH4 machines as standard equipment and has operated with them on soft fields, which would have been impossible with standard DH4 wheels and tires. Nor did these larger tires and wheels slow down speed to any noticeable extent. There seems to be almost no reasonable limit to the size of wheels which may be advantageously used.

84 In a large number of forced landings, where damage results, the front end of the machine only suffers, such as the propeller and the radiator and possibly the front end of the fuselage when the plane "noses over." This necessitates laying up the whole plane during the repair operation. To obviate this, some designers are using a separate quick-detachable nose which carries the engine. The advantages of such construction are too obvious to warrant discussion.

85 It will no doubt be found desirable in commercial machines to use unit construction as much as possible, so that replacements may be quickly made and the damaged unit repaired at leisure.

86 Experience of the Mail Service has shown that other engineering details are susceptible of considerable improvement.

Gasoline tanks have heretofore been built of such shape as to fill completely some available space in the fuselage, resulting in shapes having large, flat areas without the necessary strength. Cylindrical tanks with bumped heads would be much stronger and less troublesome and should be used if possible. They should be so installed as to be easily removed and replaced. It has been found that under certain conditions (possibly gasoline with high sulphur content) the lining of both copper and terne-plate tanks and of copper tubing is attacked and passes along to the carburetor as a fine, flocculent deposit which may cause serious interference with carburetor action. Ample strainer capacity should be located at both tank outlet and carburetor inlet and strainers made so as to be easily cleaned without breaking pipe connections. The U. S. Navy has found that aluminum gasoline tanks and piping are not attacked in the above manner and are preferable to copper.

87 It has been well demonstrated that a fuel-feeding system using air pressure on the tanks is neither reliable nor safe. The best results have been obtained with a positive gasoline-pump system using an engine-driven pump embodying automatic by-pass regulation for maintaining constant discharge pressure.

88 Instrument and ignition leads in the Army DH machines, such as tachometer driveshafts, thermometer and pressure tubes, shutter controls, ignition cables, etc., were installed in a most inaccessible manner inside the fuselage. In the rebuilt mail planes these leads are all installed in a conduit on the outside of the fuselage where inspection and replacement may be easily accomplished.

89 Intelligent engineering attention to the many details of airplane design will be well repaid in more efficient operation and decreased cost. We do not need to seek for new or radical types with which to run a commercial service. The present types, properly engineered, will deliver a most satisfactory service if handled by a properly organized and operated personnel.

90 It may be interesting to visualize the difference as applied to commercial work between the rebuilt military planes used by the Mail Service and a plane designed especially for commercial work, both planes being equipped with 400-hp. Liberty 12 engines. It should, of course, be borne in mind that the mail plane is capable of carrying more than the 400-lb. mail load, which is, however, all that can be put in the space available in these machines.

91 Some idea of what a commercial plane, using a 400-hp. engine, can do, is given in the following data, which is based on

accurate knowledge of proved performance. Such a plane having a lift-drift radio of 9 will not weigh over 5 lb. per sq. ft. of wing area. Beside fuel and oil for a 4-hour flight, together with a pilot and other accessories, such a plane will carry an actual pay load of 3000 lb. at a cruising speed of 100 miles per hour with the engine throttled down so as to be developing only two-thirds of its maximum 400 hp. Such a plane will cost no more to operate than the mail planes discussed herein.

CONCLUSIONS

92 While both the design and operating methods are susceptible of improvement, it is believed that the following are warranted from our present knowledge and experience:

a The number of planes required for a given service is about one (1) for each 250 miles of scheduled flight

b The number of spare engines required is about 50 per cent of the total number of planes

c The total operating cost should not exceed 70 cents per plane-mile for single-engine planes of not over 400 hp.

d Scheduled flights over reasonably long distances can be at least twice as fast as the fastest scheduled trains, flying in daylight only, and at least three times as fast if night flying is done

e Night flying can be successfully accomplished with reliable power plants and a proper equipment of airdrome beacons and landing flares

f The most important and desirable improvement in present practice is the use of efficient commercial planes equipped with a thoroughly reliable power plant

g The United States is admirably adapted to commercial airplane service because: (1) the large area permits long flying distances; (2) large centers of population well distributed over this area should supply ample business, and (3) fast transportation of both goods and passengers is more in demand than in any other country in the world

h The development of a large commercial service will furnish a reserve of planes, pilots, mechanics and other highly skilled men which would undoubtedly be the most valuable and effective part of our national defense.

93 The author desires to acknowledge with thanks the assistance given by Mr. C. F. Egge, General Superintendent, and Mr.

Chas. I. Stanton, Assistant General Superintendent of the Air Mail Service, in gathering the necessary data regarding the operation of the Air Mail Service; by Mr. C. A. Parker, formerly of the Air Mail Service, in furnishing valuable information regarding plane and engine maintenance; and by Mr. Chas. B. Kirkham, Aeronautical Engineer, in contributing valuable suggestions regarding possible improvements affecting the design and performance of commercial machines.

DISCUSSION

R. B. C. Noorduyn.[1] The statement was made to the writer in Washington that the carriage of mail would gladly be handed over to private transportation companies provided they could show satisfactory prospects of rendering good service, but that the Congressional appropriations did not allow the payment of a higher rate per pound for aerial transport than by any other means of transportation, such as the railroads. Air transportation is faster, and of course more pay will have to be handed over for that class of service.

In the past there was a time when the mails were not transported by railroad, but by coaches. When the transportation of mail was handed over to the railroads, a special law was passed allowing an increased payment of 25 per cent. When the airplane becomes more generally recognized as a faster vehicle some such amendment will have to be made to enable more money to be paid for the increase in speed and the improvement in public service which is obtained. If the policy of cheapness were adhered to, mail would still be carried by sailing ships rather than by modern steamships.

E. P. Warner. A short time ago Mr. Paul I. Zimmerman, engineer in charge of the Airplane Division of the Aeromarine Plane & Motor Company, wrote as follows of his company's experience in commercial operation. "Our experience with commercial flying boats is limited to a period which extends back to the signing of the armistice, and the early part of which time was spent entirely in experimenting with our boats and motors. Finally we decided that the Navy surplus flying boats were adaptable for commercial purposes and until these boats had been sold, it would be impracticable for anyone to start an extensive construction program.

[1] Netherland Aircraft Mfg. Co., 286 Fifth Ave., New York, N. Y.

"We, therefore, purchased an F–5–L and put it in operation, converted as we thought advisable. Particular pains were taken to see that the comfort of the passengers was provided for in the extreme. The interiors of the cabins are lined with polished veneer and give much the same appearance as the interior of a Pullman car. Curtains at the removable glass windows and comfortable upholstered wicker chairs add much to the comfort of the passengers. Five more of these boats were afterwards purchased and put in operation about November 1920. One of these boats was wrecked while anchored in Havana harbor; the remaining five are still in operation, one of which, the *Santa Maria*, has been operated for more than 350 flying hours and as yet shows no signs of wear, which would indicate that the boat can be used for two or three more years.

"The bottoms of the first HS–2 boats converted depreciated very rapidly. We have found it necessary to change these bottoms after about 215 hours flying time. The *Santa Maria* bottom, however, I understand, shows no signs of depreciation even after 350 hours of operation under all kinds of climatic conditions, this boat having made one hundred trips between Key West and Havana and has since been flown from Key West, via New York, Chicago, St. Louis, back to Pensacola on her way to Key West for the winter season operations.

"The covering on the wings, particularly directly in the slip stream of the propellers, depreciates very rapidly, it having been necessary to change this portion of the covering on the *Santa Maria*. A new wing covering will probably be required on the *Santa Maria* before she starts operating this winter. The expense of this, however, is not very great as compared with the revenue produced, and considering the fact that the *Santa Maria* has not been under cover since its launching last November, we are not surprised that the wing covering has depreciated.

"The motors are overhauled after 110 hours operation. We have not as yet had occasion to decide that any of the motors have worn out and are ready for the junk heap. The overhauled motors seem to be just as good if not better in some instances than the new ones, which would indicate that probably the motors would be good for ten or twelve overhaulings before being ultimately discarded, although this, of course, is only a guess and cannot be taken as indicating the life of a motor.

"All in all, our experience with operating commercial flying boats for the past year seems to indicate that a life of three or four

years may be readily assumed for the hulls and wings and that 800 to 900 hours are to be expected from the motors."

JOSEPH A. STEINMETZ. This is a very remarkable paper and the writer wants to go on record as stating that it is of real significance. Perhaps eighteen months ago at the Engineers' Club at Philadelphia a city planning conference met, and one hundred engineers and architects attended who did city planning as their professional work. One of the final results of their conference was the endorsement of a motion that it shall be hereafter the policy in engineering relating to city planning, to incorporate, in addition to parks and railroad facility areas, well designed aircraft landing fields and water areas for hydroavions.

The Acro Club of Pennsylvania, the oldest organization of its kind, has sent out recommendations as to air landing fields, and was helpful in putting on the air map of the United States numerous landing fields for smaller cities. These are not often called landing fields at first discussion, but the cities are induced to set aside an area for a fair ground or exhibition field. Then the representatives of the Air Associations go over the suggested plans and locate the buildings in proper places so as to leave a large open air field in the middle. Later the cities realize their great wisdom and foresight in creating this available field for an air port for their district.

The writer believes that China will be among the great nations in the larger uses of the airplane. In five years from now the writer believes that several thousand airplanes will be working in China; and largely because their country roads are not good and quick official transport and mail service is urgently needed.

In Europe there is an air route to Africa, French Morocco and the daily service between Paris and London is liberally patronized.

In this country is the trip from Key West to Cuba. In the United States of Colombia there are two routes, one up the Magdalena River to Bogota, making it possible to cross the fever zone there without danger or risk. In the Belgian Congo in Africa a regular air express serves for mail, passenger and valuable merchandise and supplies.

These few examples serve but to show us the fragments of the great world mosaic now in the making, of the interlinking of the air routes.

E. E. ALDRIN told of an interesting experience of his in travelling between Paris and London. Leaving Paris by train at 11.30 in

the morning, he was detained over-night at Calais owing to the rough sea in the Channel, so that he did not reach London until the following day at noon. On the return trip by airplane ten days later, the distance was covered in three hours, the plane being fifty minutes overdue because of heavy fog and a head wind.

AUGUSTUS POST[1] spoke in endorsement of the paper and the discussion of it. He said, in part: "It seems as if there was a real use for the planes among the inhabitants of the Middle West. Many men have said they were taking professional men and doctors around in their planes when the roads were impassable, and I think besides the great lines that will be established before long, that there will be many individual planes used in different sections of the country to carry the professional man to these outlying districts where other transportation is almost impossible at times. It has been said that if you had three doctors on your list, you could get on with a jitney plane by taking them out when required — that is when it was necessary to go out to locations where the roads were very bad."

THOMAS MANLEY said in part: "At present we have to keep thoroughly in mind that everything is to be sold. The people are not going to come up and ask to have the airplane delivered on a silver platter, and then hand the money in change for it on a silver platter. The uses we have must be worked out thoroughly, as they have done with the work in British Columbia and in the Congo. The big thing seems to be to study out the lines of least resistance to sell what we have in those lines, and to keep thoroughly before us the fact that it is the thin end of the wedge that we must work on all the time, and not try to drive the dull end of the wedge first. When we try to put on aerial lines in competition with highly developed trans-continental lines, where capital is great, invested interests are large, and where cities have been built around those systems of transportation, we find great difficulties. We must consider the value of time. If time is not of much value it is not worth saving. If people in rural communities are not putting high value on time, it is not a good slogan for us to use in the rural communities. If you will use the word "inconvenience," that is another thing. We save them inconvenience. I think that has been one of the great troubles in some of the operations. The big thing we have to do is to study and analyze thoroughly the situation, and

[1] 297 Madison Ave., New York, N. Y.

where these machines can be used in the present form and development. We must try to work out the different factors and bear in mind the fact that we must sell and keep selling the stuff we have right along."

THE AUTHOR, in closing the discussion, said: "One phase of the situation which we must not lose sight of is the great promise of commercial success in carrying *goods*, rather than in the transportation of *passengers*. The promised revenue is greater and the hazard very much less. The profit of flying airplanes probably lies in flying considerable distances. One or two facts may show what may be done in this country for a commercial line of airplanes. In the first place, the amount of express matter flowing between New York and Chicago daily is eighteen trainloads of an average of ten cars each. If we got a fraction of one per cent of that amount it would constitute a comfortable airplane business. On the authority of one of the directors of a Reserve Bank in New York City, there is an average daily flow of $500,000,000 in currency and negotiable paper between the various Federal Reserve Banks of the United States. The saving of one day's interest alone on that amount of money would be well worth paying for.

"If you can fly goods at fifty cents per ton-mile, a thousand-mile trip will cost $500. If you carry a ton load at $2.00 per pound you get $4000 for the trip. For a half load you get $2000 for the trip. Even at a lesser rate there is still an inviting profit. We may soon obtain legislation which will permit us to put a special stamp on letters taken by airplane. If only a five-cent stamp is put on in addition to the two-cent stamp, there being an average of forty letters in a pound of first-class mail, the revenue will be $2.00 a pound, and for a fifty per cent load you have $2,000 per trip. We do not want passengers now. With enough express business or other high-class matter we get away from the risk of carrying passengers and have a more profitable load to carry. I think the cultivation of this class of business will be much easier than that of carrying passengers.

"We have great distances to fly over and the country has been somewhat educated to transporting tremendous amounts of money, mail and express matter through the express companies. Carry a small fraction of that high-class business and it will be a relatively large airplane business and with sufficient profits for any commercial airplane company."

No. 1809

A STUDY OF THE ELASTIC PROPERTIES OF SMALL-SIZE WIRE CABLE

By R. R. Moore,[1] Dayton, Ohio

Non-Member

In aircraft work use is made of several varieties of small-diameter steel cable knowledge of the physical characteristics of which is important for efficient design. The present paper gives results of a series of tests carried out at McCook Field, Dayton, Ohio, in which it is shown that the modulus of elasticity of small-sized wire aircraft cable varies from 15,000,000 to 28,000,000, depending upon the size and type of the cable. The maximum difference between moduli of specimens of the same size and type of cable may be as high as 3,000,000.

It was also found that the modulus of elasticity may be raised by loading the cable below the elastic limit, and that resting the cable does not seem to have any definite effect on the modulus. The elastic limit may be raised by loading the cable a little beyond this point, the maximum increase obtained amounting to 63 per cent of the original elastic limit.

THE development of aircraft has brought into use several varieties of small-diameter steel cable, a knowledge of the physical characteristics of which is important for efficient design. Such information becomes vitally important to engineers who are concerned with the development of new types of aircraft and the advancement of the science of aviation in general. This is the case at the Engineering Division of the Air Service, located at McCook Field, Dayton, Ohio. The subject was first brought to the attention of the Material Section by a request from the Design Section for stress-strain curves on various sizes of the 19-wire non-flexible type of aircraft cable. After completing these tests the investigation was extended to cover all the available types of cable used in airplane construction.

2 Engineering handbooks do not quote any value for the elastic limit of wire cable, and for the modulus of elasticity they give

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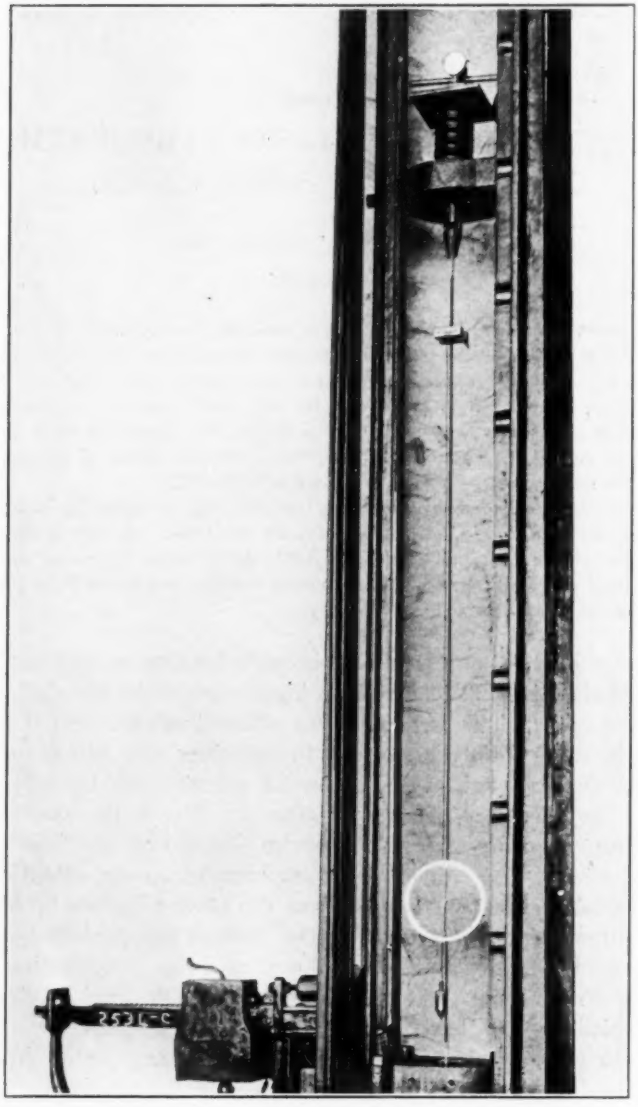


FIG. 1 COMPLETE SET-UP OF APPARATUS WITH EXTENSOMETER ATTACHED

a value of about 12,000,000. This value, however, refers to wire ropes with hemp centers and for sizes greater than $\frac{3}{8}$ in. In aircraft work we are mostly concerned with ropes made of high-strength steel wire in sizes less than $\frac{3}{8}$ in. and types which have either a wire or wire-strand center. The value for the modulus of this type of cable is considerably greater than 12,000,000.

3 While the determination of ultimate strength is a comparatively simple matter, the accurate measurement of elongation necessary in determining the elastic limit and modulus of elasticity

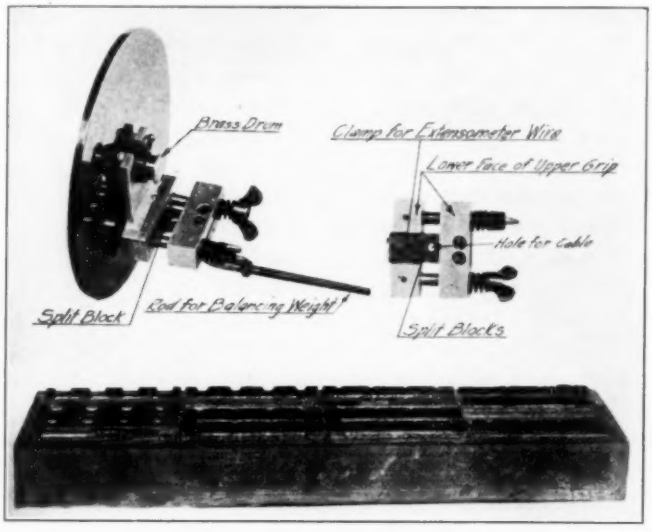


FIG. 2 EXTENSOMETER CLAMP

presents several difficulties. In the first place, the fact that the cable is built up of so many small wires and strands makes it difficult to locate accurately a definite gage point and attach an instrument there without its shifting in position. It is evident that the usual type of extensometer with pointed screws for grips is of no use for this work. In the second place, a cable twists when under load, so that any rigid instrument cannot be used.

4 The development of an instrument capable of measuring accurately and to a small degree the elongation of a cable under load was necessarily the first step. After having accomplished this, the following theoretical points were taken up: the determination

of the elastic limit and modulus of elasticity of the different types of aircraft cable in use; the determination as to whether the modulus of elasticity is changed after the cable has been loaded several times; and the determination as to whether the elastic limit can be increased by loading the cable a little beyond this point.

5 Tests were made on a 20,000-lb. Olsen structural material and airplane parts testing machine which was previously calibrated and found to be in good adjustment. However, in several cases

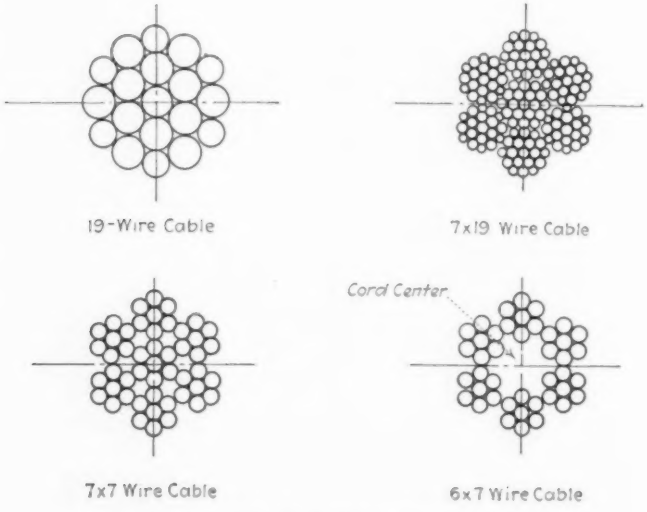


FIG. 3 TYPES OF CABLE CONSTRUCTION

the measurement of load was obtained from the deflection of an accurately calibrated spring placed on the upper head of the machine and arranged so as to take the load applied to the cable. The deflection of the spring was measured with an Ames dial reading to 0.001 in. It was found that this method gave more consistent results because it was considerably more sensitive to small variations in load than the lever system of the testing machine. In fact, some such arrangement is absolutely necessary in testing the small cables, particularly the $\frac{1}{32}$, $\frac{1}{16}$, and $\frac{3}{32}$-in. sizes, in order to obtain the elastic limit and modulus with any degree of accuracy. To obtain the breaking load, however, the spring must be removed and the load measured on the machine. A view of the complete set-up with extensometer attached is shown in Fig. 1.

6 Elongations were measured over a gage length of 50 in. in order that unit elongations might be obtained with considerable accuracy. With the extensometer shown it was possible by using the vernier to read to 0.0002 in. Without the vernier the smallest direct reading is 0.002 in., so that if this be divided by the gage length of 50 in., the unit elongation which is absolutely dependable is $0.002/50 = 0.00004$ in., which is nearly as fine as most extensometers used for obtaining the modulus on solid bars will read.

7 The instrument used was very similar to the familiar wire-wound dial type of extensometer. The greatest difficulty was in attaching the dial to the cable at the gage point. This was accomplished by attaching the dial to a specially devised clamp as shown in Fig. 2. The clamp consisted of two small aluminum castings, each slotted to receive half of a split block ($\frac{1}{2}$-in. by $\frac{3}{4}$-in. by $\frac{1}{16}$-in. thick), this block having a hole drilled centering on the dividing line to receive the cable. The split block was forced together by means of a wing nut on the threaded end of each of two bolts, one end of each of which was set tight in one casting and the threaded end extended through two holes in the other casting and projected beyond sufficiently to take the wing nut. The dial itself was about six inches in diameter and graduated into 1000 divisions. A needle containing the vernier was mounted on a shaft extending through the dial, the other end of the shaft holding a small brass drum. The drum was so proportioned to the dial that one revolution of the dial was equivalent to a 2-in. stretch on the cable or one small division on the dial = 0.002 in.; using the vernier this could be read to 0.0002 in. At the other gage point, a second clamp was attached similar to the one holding the dial. To this clamp one end of a fine magnet wire was attached; the wire extended down the cable and around the brass drum, the loose end having a light weight attached to it. It is clear, then, that as the two gage points moved apart the wire turned the drum and shaft upon which the needle was mounted, thus recording directly on the dial the elongation of the cable.

8 This method of attaching the instrument to the cable proved quite satisfactory, for it was easy to set in place and it held firmly. The only adjustment in the instrument in order to adapt it to different sizes of cable was a new pair of split blocks with a hole drilled to take the cable. A complete set of these blocks was made and filed as shown in Fig. 2, so that no time was lost in adapting the instrument to the different-sized cables. It was found later on that

split blocks made of fiber served very satisfactorily and were much lighter.

9 As mentioned in Par. 3, the second difficulty in measuring elongation is due to twisting of the cable. As the cable twists the clamps attached to the cable rotate about the center of the cable as an axis. Both clamps do not move in the same direction, but in opposite directions. It is clear, then, that the point where the

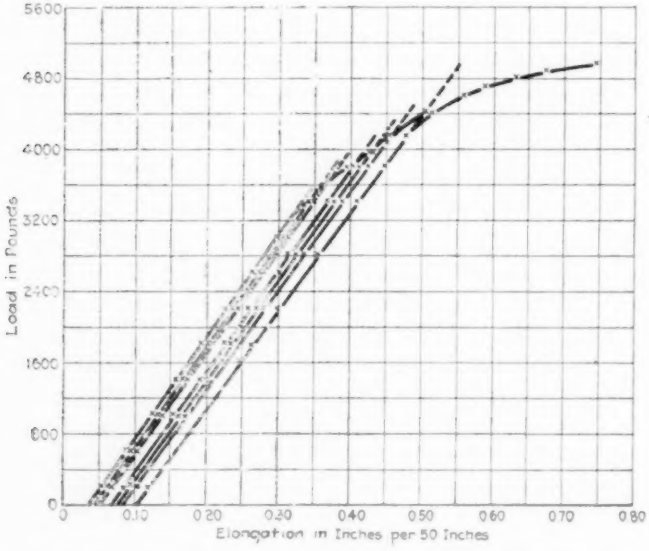


FIG. 4 RISE IN ELASTIC LIMIT OF $\frac{3}{16}$-IN. 19-WIRE NON-FLEXIBLE STEEL-WIRE CABLE

extensometer wire is attached to the upper clamp is no longer directly over the point of contact between the wire and the brass drum of the dial. The length of the wire between these two points has increased by an amount equal to the difference between the length of the hypotenuse and the longer side of a right triangle. In order to reduce the error occurring from this twisting, the wire was placed as near the cable as possible and hardly exceeded $\frac{1}{2}$ in. from the center of the cable at any time. As the maximum twist noted did not exceed 90 deg., the error incurred in the modulus due to this error of elongation is negligible.

10 Specimens were 6 ft. long overall and were socketed at

each end. The individual wires were frayed out at each end of the specimen for a distance of about two inches; they were then cleaned by dipping in a solution of zinc chloride and coated with the solder used in the sockets. The frayed ends were then set in the ordinary closed socket and soldered there with solder composed of 50 per cent lead, 25 per cent tin, and 25 per cent antimony, which alloy has a melting point of 625 deg. fahr. This alloy was selected because its

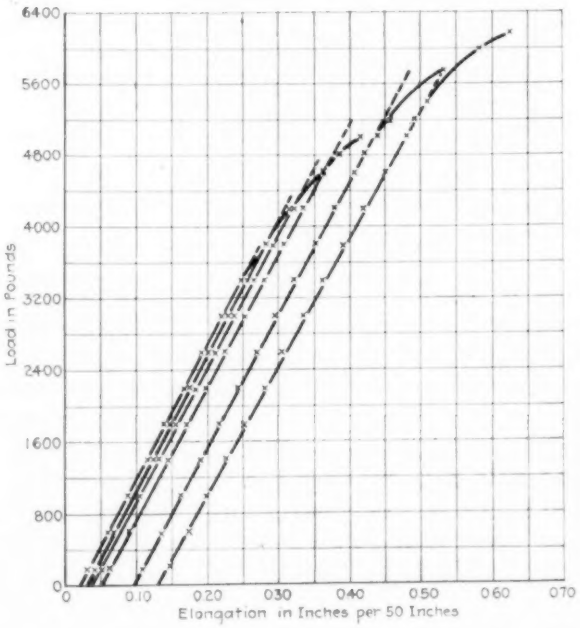


FIG. 5 RISE IN ELASTIC LIMIT OF $\frac{7}{32}$-IN. 19-WIRE NON-FLEXIBLE STEEL-WIRE CABLE

melting point is low enough to have little effect on the hardness of the steel wire, and it is strong enough to hold the heaviest cables tested. In fact, in no case did the cable break in the socket. The sockets used were the standard commercial "closed socket," the outer surface of which was turned up on a lathe so that it fitted snugly into the tapered hole in a split block. The split block was made to fit the holes in the head of the testing machine. This method made it possible to set the specimen up quickly and with good alignment.

11 Cables of four different types of construction and one made of phosphor-bronze wire were tested, the types and sizes tested being given in Table 1.

TABLE 1 TYPES AND SIZES OF CABLES TESTED

Type	Material of Wires	Protective Coating on Wires	Diameter of Cable
19 wire	Steel	Tinned	$\frac{1}{32}$,[1] $\frac{1}{16}$, $\frac{3}{64}$, $\frac{1}{8}$, $\frac{5}{32}$, $\frac{3}{16}$
7 by 7	Steel	Zinc-plated	$\frac{1}{16}$, $\frac{3}{32}$, $\frac{1}{8}$
7 by 7	Steel	Tinned	$\frac{1}{8}$
6 by 7	Steel	Zinc-plated	$\frac{3}{64}$
7 by 19	Steel	Tinned	$\frac{1}{8}$, $\frac{5}{32}$, $\frac{3}{16}$, $\frac{5}{32}$, $\frac{1}{4}$
7 by 19	Phosphor-bronze	None	$\frac{1}{8}$

[1] This cable has only 7 wires because of its small size

12 The 19-wire cable is made up of 18 wires twisted around a wire core, the wires taking the direction of a left-hand screw.

13 The 7 by 7 cable is made up of seven strands of seven wires each, six of these strands being twisted around the seventh strand as a core, the strands being twisted in the direction of a right-hand screw. The strands are composed of seven wires, six of which are wound around the seventh as a core, the direction being that of a left-hand screw. A cable wound in this manner, that is, the individual wires twisted in the opposite direction from the twist of the strands, is said to be "regular lay" cable.

14 The 7 by 19 cable is made up of seven strands of 19 wires each. The method of construction is the same as for the 7 by 7 cables.

15 The 6 by 7 cable is composed of six strands of seven wires each, the six strands being wound around a cord center. The strands are composed of seven wires, six of which are wound around a wire center. The cord used for the center is cotton twine of cable construction, there being three strands each consisting of eight yarns of about No. 20. The phosphor-bronze cable is of the same construction as the regular 7 by 19 cable.

16 A chemical analysis of several wires taken from the phosphor-bronze cable gave the following composition: tin, 3.77; copper, 95.60; zinc, 0.34; phosphorus, 0.28. A chemical analysis of several wires from the steel cables gave the following composition: carbon, 0.64; manganese, 0.55; phosphorus, 0.033; sulphur, 0.033.

17 In Fig. 3 is shown the arrangement of the wires and strands in the four different types of cables.

18 Each sample of cable was cut from the reel in a single 30-ft. length. From this piece five 6-ft. specimens were taken for test. In this manner it was assured that all specimens of the same

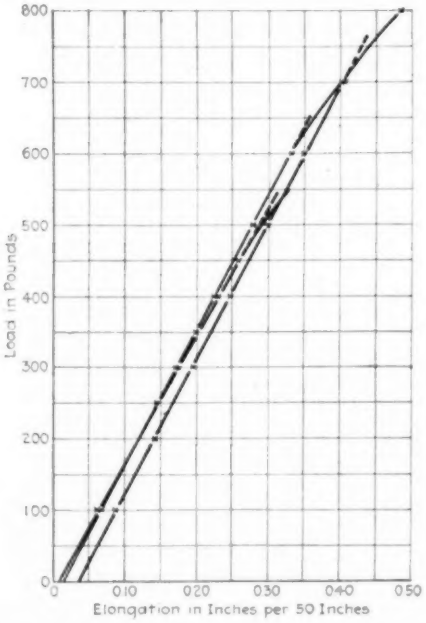


FIG. 6 RISE IN ELASTIC LIMIT OF $\frac{3}{32}$-IN. 7 BY 7 NON-FLEXIBLE STEEL-WIRE CABLE

size and type came from the same material. Four specimens of each were prepared for test, the fifth being held in reserve.

19 The four specimens of each size and type were run as follows: Three runs on each of two specimens, taking elongation readings at equal increments of load up to near the breaking point, three runs on each of the other two specimens, taking elongation readings at equal increments of load up to a point somewhat below the elastic limit, then one run above the yield point, rest for two days, then two runs up to near the breaking load. The purpose of this arrangement was to determine whether the modulus was affected by loading below the elastic limit or above the yield point, or by rest after loading.

20 The cross-sectional area of the cable was found by taking the sum of the areas of the individual wires. These areas were computed from the diameters of the wires measured to thousandths with micrometer calipers.

21 Stress-strain curves for computing the modulus and locating the elastic limit were plotted on cross-section paper with a scale of 1 in. equal to 0.01 in. actual elongation in 50 in. The vertical scale was made so as to give the curve a slope of about 45 deg. with the horizontal. This large-scale plotting showed the readings of

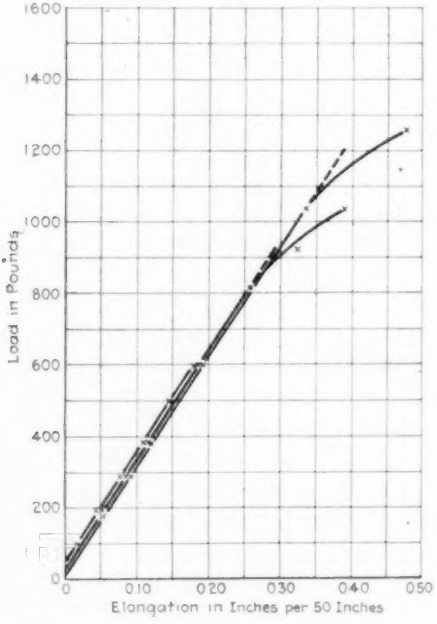


FIG. 7 RISE IN ELASTIC LIMIT OF $\frac{3}{4}$-IN. 6 BY 7 STEEL-WIRE CABLE

the extensometer to be very consistent, indicating that the cable was stretching uniformly and also twisting uniformly.

22 The values for elastic limit were obtained by selecting the highest point on the straight-line section of the stress-strain curves. Strictly speaking, this is not the elastic limit but the proportional limit. The American Society for Testing Materials defines these two points as follows:

Elastic Limit is the greatest load per unit of original cross-section which does not produce a permanent set. (This determination is rarely made in the commercial testing of materials.)

Proportional Limit is the load per unit of original cross-section at which the deformations cease to be directly proportional to the loads.

23 However, as the majority of practical designing engineers use the term "elastic limit" to designate what is actually the propor-

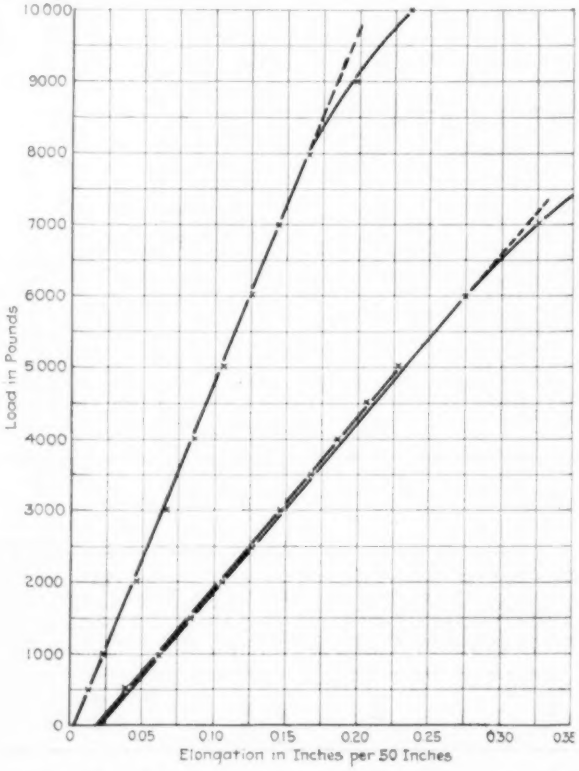


FIG. 8 RISE IN ELASTIC LIMIT OF $\frac{1}{2}$-IN. 7 BY 19 FLEXIBLE STEEL-WIRE CABLE

tional limit, we have also used it in that sense in order to avoid confusion.

DISCUSSION OF RESULTS

24 A complete summary of the results of tests is given in Tables 2 to 6. The entire work consists of tests on 47 specimens of wire cable which comprised 103 stress-strain runs.

TABLE 2 19-WIRE NON-FLEXIBLE STEEL-WIRE CABLE

1 Size	2 Area	3 Elastic limit, lb.	4 Break-load, lb.	5 Modulus of elasticity	6 Spec. No.	7 Treatment	8 Increase in Modulus due to loads below elastic limit	9 Increase in Modulus after loading above Yield Point	10 Average first-run modulus
$\frac{1}{32}$ [1]	0.0008023	135	180	24,420,400	3	Loaded near breaking point			24,380,000
				24,420,400		Loaded near breaking point			
		110	180	24,344,100	4	Loaded to elastic limit			
				24,000,000		Loaded to elastic limit			
				23,932,100	38	Below elastic limit			
				24,480,800		Below elastic limit			
				23,932,100		Below elastic limit			
				24,480,800		Loaded near breaking point			
				23,590,900		After two days' rest			
				23,030,700		After two days' rest			
				24,825,600	39	Loaded above yield point			
				25,074,100		Loaded above yield point			
				25,047,800		Loaded above yield point			
$\frac{1}{16}$	0.0020588	250	600	25,281,400	37	Loaded below elastic limit	500,000		25,281,400
				25,799,600		Loaded below elastic limit			
				25,799,600		Loaded below elastic limit			
				25,799,600		Loaded near breaking			
				26,369,100		After two days' rest			
				25,386,000		After two days' rest			
$\frac{7}{64}$	0.0068823	800	1800	25,641,200	12	Loaded near breaking point		250,000	25,641,200
				25,398,000		Loaded near breaking point			
$\frac{1}{8}$	0.008878	1200	2350	27,728,200	2	First run	75,000	340,000	27,728,200
				27,802,000		After loading 4 times below E.L.			
				27,389,200		After loading several times above Y.P.			
$\frac{5}{32}$	0.017224	1800	3840	24,837,900	3	First run	700,000	1,100,000	24,870,000
				25,509,000		After loading 4 times below E.L.			
				25,918,900		After loading several times above Y.P.			
	0.017224	1800	3880	24,912,200	4	First run	600,000	600,000	
				25,520,200		After loading 3 times below E.L.			
				25,455,700		After loading several times above Y.P.			
$\frac{3}{16}$	0.020619	3200	4970	26,159,100	5	First run	1,600,000	1,450,000	26,195,000
				27,779,800		After loading 7 times below E.L.			
				27,608,500		After loading several times above Y.P.			
	0.020619	2800	5370	26,231,400	7	First run	1,550,000	1,500,000	
				27,793,100		After loading 3 times below E.L.			
				27,793,100		Next run			
$\frac{7}{32}$	0.028761	3200	6570	25,440,970	6	First run	300,000	800,000	25,550,000
				25,767,780		After loading 4 times below E.L.			
				26,265,800		After loading several times above Y.P.			
	0.028761	4200	6480	25,660,000	8	First run	100,000	100,000	
				25,755,000		After loading 3 times below E.L.			
				25,755,000		Next run			

[1] This is a 7-wire cable.

E.L. . . . Elastic limit.

Y.P. . . . Yield point.

TABLE 3 TESTS ON 7 BY 7 FLEXIBLE STEEL-WIRE CABLE

1 Size and coating	2 Area	3 Elastic limit, lb.	4 Breaking load, lb.	5 Modulus of elasticity	6 Spec. No.	7 Treatment	8 Average first-run modulus	9 Max dif. between 1st run mod. of specimens of same cable
3/8 Zinc	0.002198	225	580	19,051,900	9	Single run to near breaking	19,399,000	700,000
		250	600	19,746,400	10	Single run to near breaking		
3/8 Tinned	0.0038427	400	1170	19,936,700	5	Loaded to elastic limit	20,801,620	2,000,000
				22,438,600		Loaded to near breaking		
		400	1130	20,182,900	6	Loaded to near breaking		
				22,075,000	35	Loaded below elastic limit		
				22,993,100		Loaded below elastic limit		
				22,993,100		Loaded below elastic limit		
				22,993,100		Loaded above yield point		
				23,936,700		After two days' rest		
				23,882,800		After two days' rest		
				21,011,900	36	Loaded below elastic limit		
				22,366,200		Loaded below elastic limit		
				22,893,800		Loaded below elastic ilmit		
				23,009,800		Loaded above yield point		
				23,477,100		After two days' rest		
				23,590,400		After two days' rest		
3/8 Zinc		450	1000	22,121,000	7	Single run to near breaking	21,736,000	800,000
		400	1000	21,351,200	8	Single run to near breaking		
3/8 Zinc-plated	0.0123985	1300	3000	20,116,900	17	Loaded above yield point	19,504,500	1,300,000
				23,408,800		Loaded above yield point		
				23,408,800		Loaded above yield point		
		1400	3100	18,892,100	18	Loaded above yield point		
				22,753,500		Loaded above yield point		
				22,560,800		Loaded above yield point		

TABLE 4 TESTS ON 6 BY 7 STEEL-WIRE CABLE (CORD CENTER)

1 Size and Coating	2 Area	3 Elastic limit, lb.	4 Breaking load, lb.	5 Modulus of elasticity	6 Spec. No.	7 Treatment	8 Average first-run modulus	9 Max. dif. between 1st run mod. of specimens of same cable
$\frac{9}{64}$ Zinc-plated	0.0072967	876	1660	17,037,800	15	Loaded above yield point	17,988,700	2,760,000
				21,709,630		Loaded above yield point		
				21,468,900		Loaded above yield point		
		820	1680	17,020,900	16	Loaded above yield point		
				21,158,700		Loaded above yield point		
				21,091,600		Loaded above yield point		
				19,781,800	40	Loaded below elastic limit		
				20,817,500		Loaded below elastic limit		
				20,817,500		Loaded below elastic limit		
				21,065,800		Loaded above yield point		
				20,421,100		After two days' rest		
				21,071,400		After two days' rest		
				18,114,400	41	Loaded below elastic limit		
				19,013,300		Loaded below elastic limit		
				19,013,300		Loaded below elastic limit		
				19,013,300		Loaded above yield point		
				19,850,600		After two days' rest		
				21,013,800		After two days' rest		

TABLE 5 TESTS ON 7 BY 19 EXTRA FLEXIBLE STEEL-WIRE CABLE

1 Size and coating	2 Area	3 Elastic limit, lb.	4 Breaking load, lb.	5 Modulus of elasticity	6 Spec. No.	7 Treatment	8 Average first-run modulus	9 Max. dif. between 1st run mod. of specimens of same cable
$\frac{1}{8}$ Tinned	0.0073715	1260	1910	15,054,700	13	Loaded above yield point	16,627,300	3,200,000
				20,057,390		Loaded above yield point		
				20,019,550		Loaded above yield point		
		1140	2135	18,318,550	14	Loaded above yield point		
				23,319,170		Loaded above yield point		
				23,153,890		Loaded above yield point		
			2130	15,728,400	33	Loaded below elastic limit		
				21,806,100		Loaded below elastic limit		
				22,240,460		Loaded below elastic limit		
				22,257,300		Loaded above yield point		
				22,411,700		After two days' rest		
				22,371,200		After two days' rest		
			2140	17,407,620	34	Loaded below elastic limit		
				21,500,000		Loaded below elastic limit		
				21,712,000		Loaded below elastic limit		
				22,000,100		Loaded above yield point		
				21,567,300		After two days' rest		
				22,364,500		After two days' rest		
$\frac{3}{16}$ Tinned	0.012137	1250	3020	14,689,960	19	Loaded above yield point	15,604,900	1,830,000
				19,401,400		Loaded above yield point		
				19,160,200		Loaded above yield point		
		1275	3000	16,519,800	20	Loaded above yield point		
				21,063,300		Loaded above yield point		
				21,014,300		Loaded above yield point		
$\frac{1}{4}$ Tinned	0.0178207	2100	3920	15,930,300	21	Loaded above yield point	17,189,700	2,520,000
				18,034,300		Loaded above yield point		
				17,907,600		Loaded above yield point		
		1750	4500	18,449,000	22	Loaded above yield point		
				20,849,900		Loaded above yield point		
				20,724,500		Loaded above yield point		
				21,212,300		Loaded above yield point		

Continued on page 782

TABLE 5 — *Continued* TESTS ON 7 BY 19 EXTRA FLEXIBLE STEEL-WIRE CABLE

1 Size and coating	2 Area	3 Elastic limit, lb.	4 Breaking load, lb.	5 Modulus of elasticity	6 Spec. No.	8 Treatment	8 Average first-run modulus	9 Max. dif. between 1st run mod. of specimens of same cable
$\frac{7}{32}$ Tinned	0.01956	2100	5460	21,541,300	23	Loaded above yield point	21,333,600	400,000
				23,155,600		Loaded above yield point		
				22,899,200		Loaded above yield point		
		2400	5300	21,125,900	24	Loaded above yield point		
				22,960,200		Loaded above yield point		
				23,140,300		Loaded above yield point		
$\frac{9}{32}$ Tinned	0.0386847	4800	9150	17,861,800	25	Loaded above yield point	17,789,000	150,000
				20,938,240		Loaded above yield point		
				21,000,000		Loaded above yield point		
		4800	8970	17,716,500	26	Loaded above yield point		
				20,095,400		Loaded above yield point		
				20,418,600		Loaded above yield point		
$\frac{11}{32}$ Tinned	0.062253	6000	13460	16,202,300	27	Loaded above yield point	16,947,000	1,250,000
				19,818,500		Loaded above yield point		
				19,818,500		Loaded above yield point		
		6000	13610	17,040,400	28	Loaded above yield point		
				20,677,850		Loaded above yield point		
				20,677,850		Loaded above yield point		
			13950	17,088,000	31	Loaded below elastic limit		
				18,500,000		Loaded below elastic limit		
				18,500,000		Loaded below elastic limit		
				20,365,700		Loaded above yield point		
				19,782,550		After two days' rest		
				19,782,550		After two days' rest		
				17,460,250		Loaded below elastic limit		
				19,332,300		Loaded below elastic limit		
				19,400,000		Loaded below elastic limit		
				20,079,300		Loaded above yield point		
				19,892,800		After two days' rest		
				20,282,100		After two days' rest		

TABLE 6 TESTS ON 7 BY 19 PHOSPHOR-BRONZE CABLE

1 Size and coating	2 Area	3 Elastic limit, lb.	4 Breaking load, lb	5 Modulus of elasticity	6 Spec. No.	7 Treatment	8 Average first-run modulus	9 Max. dif. between 1st run mod of specimens of same cable
$\frac{7}{32}$	0.0211636	1315	2390	11,452,800	29	Loaded above yield point	11,307,000	270,000
				11,795,900		Loaded above yield point		
				11,795,900		Loaded above yield point		
		1530	2310	11,200,000		Loaded above yield point		
				12,235,700		Loaded above yield point		
				12,235,700		Loaded above yield point		
				11,187,300	42	Loaded below elastic limit		
				11,670,100		Loaded below elastic limit		
				11,670,100		Loaded below elastic limit		
				11,670,100		Loaded above yield point		
				11.746,200		After two days' rest		
				11,867,800		After two days' rest		
				11,388,800	43	Loaded below elastic limit		
				11,388,800		Loaded below elastic limit		
				11,388,800		Loaded below elastic limit		
				12,014,800		Loaded above yield point		
				11,795,900		After two days' rest		
				11,912,000		After two days' rest		

25 It will be noticed on referring to these tables that there is considerable variation between the modulus of different-sized specimens of the same type. In order to show the comparative modulus values for the different types of cable, Table 7 was prepared, selecting the same size cable from each type as near as it was possible to obtain.

TABLE 7 COMPARATIVE MODULUS VALUES FOR DIFFERENT TYPES OF CABLE

Type	Size	Average 1st Run Modulus	Average Modulus after Loading above Yield Point
19 wire	$\frac{5}{32}$	24,870,000	25,687,000
7 by 7	$\frac{3}{16}$	19,505,000	22,985,000
6 by 7	$\frac{3}{16}$	17,989,000	20,660,000
7 by 19	$\frac{3}{16}$	15,605,000	20,112,000
7 by 19 [1]	$\frac{1}{2}$	11,307,000	11,953,000

[1] Phosphor-bronze.

26 It is seen here that of the steel-wire cables the 19-wire cable has the highest modulus and the 7 by 19 the lowest. The phosphor-bronze cable has the lowest of all, as would be expected.

27 There is in several cases quite a difference between the first-run modulus obtained on different specimens of the same cable. The largest of these differences is in these $\frac{1}{8}$-in. 7 by 19 cable, specimens 13 and 14, which show a difference of about 3,200,000; and the 6 by 7 cable, specimens 16 and 40, which show a difference of 2,760,000.

28 The highest modulus obtained on any cable is 27,793,100, which is shown in Table 2 on the $\frac{3}{16}$-in. 19-wire cable, specimen No. 7. This value was obtained after loading below the elastic limit. It should be remarked here that this value is more than twice as large as the value 12,000,000 which is commonly quoted for wire rope.

29 The increase in modulus due to repeated loading is shown in Table 8, columns 5 and 6. Notice should be taken of the fact that in column 6 the increase is not in all cases entirely due to a single load above the yield point, but may be due to a large extent to previous runs below the elastic limit. It can be easily ascertained to which specimens this applies by noting which specimens have their columns 4 and 5 filled in or by referring to column 7, Table 2, and column 8, Tables 3 to 6. Column 6, Table 8, was obtained by

subtracting the first-run modulus from the fifth-run modulus, or in those cases where only three runs were made above the yield point, by subtracting the first-run from the second-run modulus.

30 The largest increase in modulus is found in the $\frac{1}{8}$-in. 7 by 19 cable, which shows an increase of 6,600,000 after loading above

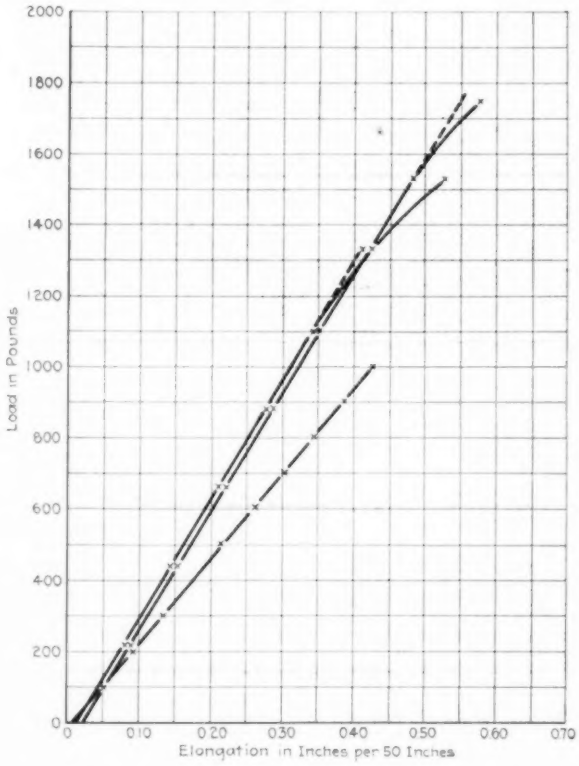


FIG. 9 RISE IN ELASTIC LIMIT OF $\frac{1}{8}$-IN. 7 BY 19 FLEXIBLE STEEL-WIRE CABLE

the yield point. It is noticeable that this maximum increase occurs on the smallest-sized cable. The phosphor-bronze cable shows the smallest increase in modulus of any of the other cables, even though it is of the same type of construction (7 by 19) as the one which showed the greatest increase.

31 While it is true that the modulus is raised a greater amount by loading above the yield point than below the elastic limit, it is

TABLE 8. EFFECT OF LOADING AND RESTING ON MODULUS

1 Type of Cable	2 Diam. of cable	3 Spec. No.	4 Increase in modulus due to single load below elastic limit	5 Increase in modulus due to loading 3 times below elastic limit	6 Increase in modulus due to loading above yield point	7 Effect of 2 days' rest on modulus after loading above yield point	8
						1st run	2d run
7 by 7 Steel-Wire Cable	3/32	5	2,500,000				
		35	900,000	900,000	1,800,000	+1,000,000	+1,000,000
		36	1,350,000	2,000,000	2,500,000	+450,000	+600,000
	5/32	17			3,300,000		
		18			3,700,000		
6 by 7 Steel-Wire Cable	3/64	15			4,600,000		
		16			4,100,000		
		40	1,000,000	1,300,000	1,300,000	−600,000	
		41	900,000	900,000	2,900,000	+850,000	+2,000,000
7 by 19 Steel-Wire Cable	1/8	13			5,000,000		
		14			5,000,000		
		33	6,100,000	6,500,000	6,600,000	+150,000	+100,000
		34	4,100,000	4,600,000	4,900,000	−450,000	+350,000
	5/32	19			4,700,000		
		20			4,500,000		
	3/16	21			2,100,000		
					2,400,000		
	7/32	23			1,600,000		
		24			1,800,000		
	9/32	25			3,100,000		
		26			2,500,000		
	11/32	27			3,600,000		
		28			3,600,000		
		31	1,400,000	3,300,000	2,700,000	−600,000	−600,000
		32	1,900,000	2,600,000	2,600,000	−200,000	+200,000
7 by 19 Phosphor-Bronze Cable	3/32	29			300,000		
		30			1,000,000		
		42	500,000	500,000	700,000	+50,000	+200,000
		43		600,000	500,000	−200,000	−100,000

to be remembered that a very large part of this increase is due to the loading below the elastic limit. It is evident, then, that the net effect of loading above the yield point is small.

32 Specimens Nos. 31, 32, 33, 34, 35, 36, 38, 40, 41, 42, and 43 were given two days' rest after having been loaded twice below the elastic limit and once above the yield point. At the end of two days they were given two more runs, both above the yield point. A comparison of these results drawn from the data in Tables 2 to 6 is given in columns 7 and 8, Table 8. These figures were obtained by taking the difference between the last run previous to the rest and each of the runs after the rest. The plus sign indicates an increase and the minus sign a decrease in the modulus. The largest difference, which is that shown in specimen No. 41, is not substantiated by its companion specimen, No. 40. In a great many cases a decrease in modulus is also shown which would indicate that so far as these tests go there is no evidence of any definite effect of rest upon the modulus of these types of cable.

33 The final object of this investigation was to determine whether it is possible to raise the elastic limit by loading a little beyond this point. The results of runs made to show this are given in Figs. 4 to 9. These curves indicate definitely that the elastic limit can be considerably increased.

34 The amount of increase obtained on the cables tested is as given in table 9.

TABLE 9 AMOUNT OF INCREASE IN ELASTIC LIMIT OBTAINED ON THE CABLES TESTED

Type	Size	Increase in Elastic Limit, lb.	Per Cent Increase over Original Elastic Limit
19	$\frac{7}{32}$	2000	62
	$\frac{5}{16}$	1000	33
7 by 7	$\frac{3}{32}$	250	63
6 by 7	$\frac{3}{64}$	240	30
7 by 19	$\frac{1}{8}$	400	36
	$\frac{1}{4}$	2000	33

CONCLUSIONS

35 The modulus of elasticity of small-size wire aircraft cable varies from 15,000,000 to 28,000,000, depending upon the size and type of the cable. The 19-wire cable has undoubtedly the highest

modulus and the 7 by 19 cable probably the lowest. The maximum difference between moduli of specimens of the same size and type of cable may be as high as 3,000,000.

36 The modulus of elasticity may be raised by loading the cable below the elastic limit. The greatest increase is obtained on the 7 by 19 type and may amount to 6,500,000.

37 Resting the cable did not seem to have any definite effect on the modulus.

38 The elastic limit may be raised by loading the cable a little beyond this point. The maximum increase obtained amounted to 63 per cent of the original elastic limit.

No. 1810

TESTING EMERGENCY FLEET BOILERS USING OIL FUEL

By F. W. Dean, Boston, Mass.
Member of the Society

Tests upon the standard four-pass water-tube marine boiler designed by the United States Shipping Board Emergency Fleet Corporation, using coal as fuel, were reported by the present author and Mr. H. Kreisinger in a paper read at the Annual Meeting of the Society in 1919. In consequence of the introduction of oil for fuel on many of the steamships of the United States Shipping Board, it became desirable to ascertain the relative merits on several of the available mechanical atomizing oil burners for boilers. Tests were accordingly run, the boiler used having a heating surface of 2518 sq. ft., a furnace volume of 408 cu. ft., and a commercial horsepower of 435 on the basis of the marine rating of 6 lb. of water to a square foot of heating surface per hour. These tests and the results obtained form the subject of the present paper.

OIL-FUEL EVAPORATIVE TESTS

IN THE paper presented to this Society in 1919 by the author and Mr. Henry Kreisinger, on coal-fuel tests of Emergency Fleet Corporation water-tube boilers for wood ships,[1] it was stated that oil-fuel tests were being made, and in the present paper the principal results of those tests are given.

2 In consequence of the introduction of oil for fuel on many of the steamships belonging to the United States Shipping Board it became desirable to ascertain the relative merits of several of the mechanical atomizing oil burners for boilers. The tests were made under the direction of the writer assisted by Mr. Henry Kreisinger, then Fuel Engineer of the Bureau of Mines, and under the critical observation of Lieut-Commander W. R. Purnell and Lieutenant Pennycook from the Philadelphia Fuel Oil Testing Plant of the U. S. Navy. As in the coal-burning tests, assistants for chemical and

[1] Trans. Am. Soc. M. E., vol. 41, p. 623.

Presented at the Annual Meeting, December 1921, of The American Society of Mechanical Engineers.

physical work were furnished by the Bureau of Mines, as well as by the Emergency Fleet Corporation.

3 The boiler tested was the four-pass boiler illustrated on p. 627 of the former paper and here reproduced as Fig. 1. In the oil tests the bottom of the casing was dropped 12 in., and the furnace on sides and bottom was lined with brickwork about 12 in. thick, there being 2½ in. of sil-o-cel brick next to the casing. These were covered by 2½ in. of calcined brick. The sides were still further lined with firebrick 6 in. thick and the bottom covered with two courses of standard firebrick laid flat. This formed a very effective insulation, so good, in fact, that the bricks suffered accordingly. However, it was the standard lining of the Emergency Fleet and was very satisfactory in keeping down the temperature of the fire rooms aboard ship, for which purpose it was adopted. This is shown in Fig. 2.

4 The furnace volume was very small, the distance between the front and back walls was only 6 ft., and the distance from the back wall to the burner tip about 6 ft. 9 in. This short distance, without doubt, prevented the atomizing capacities of the burners from being fully developed. After every series of tests the joints between the bricks opposite each burner required pointing and sometimes the bricks over a small area required replacing.

5 The following were the dimensions of the firebox below the tubes:

Height of furnace at front	5 ft. 0½ in.
Height of furnace at back	6 ft. 9 in.
Width of furnace	11 ft. 6½ in.
Distance between front and back walls	6 ft. 0 in.
Volume of furnace	408 cu. ft.

6 The boiler was equipped with thermocouples for measuring the temperatures of the gases throughout the passes and in the uptake. At the latter level there were several thermocouples for determining the temperatures at various points, for it had been previously ascertained that the temperatures in the different parts of one horizontal plane of the uptake were not uniform and varied as much as 100 deg. fahr.

7 Means were provided for sampling the gases at various points throughout the boiler between the vicinity of the burners and the uptake. Thus the progression of the process of combustion could be ascertained. Similarly draft gages were arranged to indicate the drafts at various points.

8 Some of the dimensions of the boiler are given as follows:

Width of casing at floor level	13 ft. 4 in.
Length of casing at floor level	7 ft. 10½ in.
Height of center of steam drum above floor	11 ft. 8¾ in.
Width of water spaces in headers	8 in.
Outside diameter of tubes	3 in.
Exposed length of tubes between headers	7 ft. 7½ in.
Number of tubes connecting headers	388
Number of tubes between rear header and drum	21
Steam pressure carried	210 lb.

THE OIL

9 Mexican oil was used because it is the only oil that is usually available for steamship fuel. This oil as it comes from the wells is of about 21 gravity Baumé. About 12 per cent to 13 per cent of gasoline is removed, leaving the oil at about 16 gravity Baumé. The heating value per pound is about 18,330 B.t.u.

10 The first sample taken was found by the Bureau of Mines to have the following properties:

Specific gravity at 15 deg. cent.	0.960
Gravity, Baumé, modulus 140, at 60 deg. fahr.	15.8
Viscosity, Engler, at 65.5 deg. cent.	19.21
Calories per gram	10,179
British thermal units per pound	18,323
Water, per cent	0.00
Sulphur, per cent	3.88
Mineral matter, sand, etc., per cent	0.00
Flash point (Pensky-Martens closed tester)	72 deg. cent.
Burning point (Pensky-Martens tester opened)	116 deg. cent.
Solid at	−2 deg. cent.

Other samples, of which many were taken, differed but slightly from this.

THE BURNERS

11 As stated before, all burners tested were mechanical atomizing. This type is used at sea in order that all steam used in heating the oil may be condensed and saved for feedwater.

12 Six kinds of burners were used, but only one will be designated by name or illustrated. Notwithstanding this, it will be seen from the results of the tests that there are several good burners available. Of the others two will be omitted because one was specially

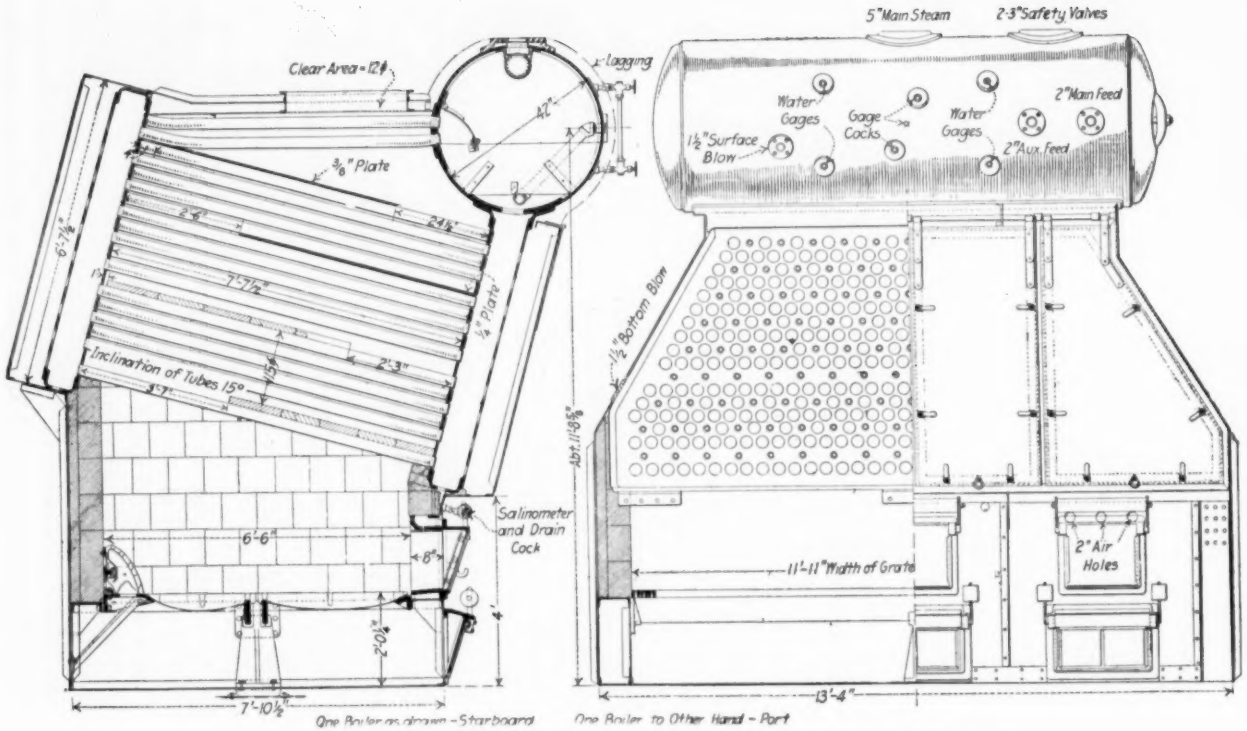


FIG. 1 FOUR-PASS STANDARD WATER-TUBE MARINE BOILER FOR WOOD SHIPS

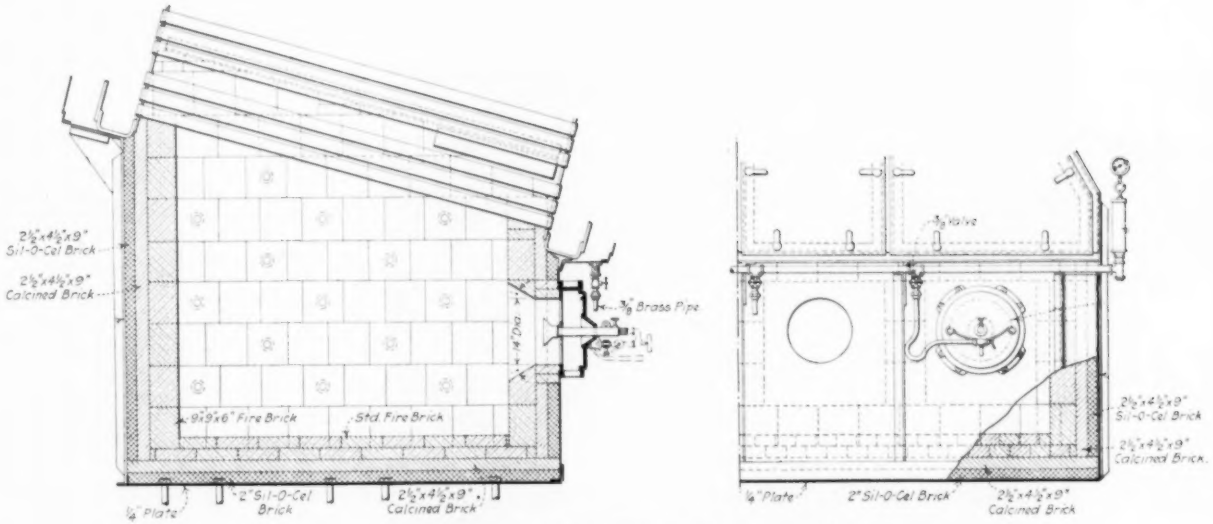


FIG. 2 ARRANGEMENT OF FURNACE OF BOILER FOR OIL FIRING

designed for use in a closed fire room, and the other was not fully developed. The burners will be designated by number, and No. 1 was the U. S. Navy Bureau (of Steam Engineering) Standard Register Burner. The reference numbers are 1, 3, 4, and 6. Fig. 3 shows the

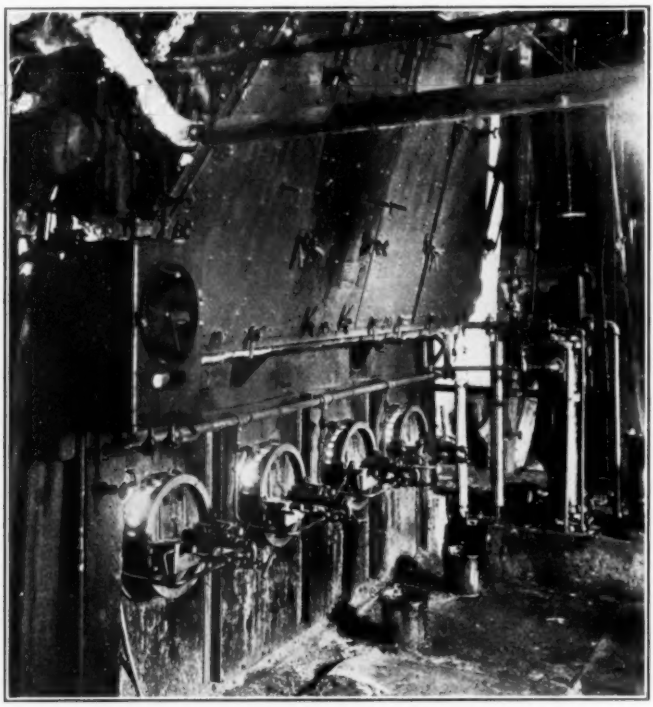


FIG. 3 BOILER EQUIPPED WITH NO. 1 BURNER

boiler equipped with No. 1 burner. Four burners were applied to the boiler, and usually they were all in use.

OTHER APPARATUS

13 The oil was supplied to the burners by either of two horizontal duplex pumps, $5\frac{1}{4}$ in. by $3\frac{1}{2}$ in. by 5 in., borrowed from the Emergency Fleet storehouse at the plant of the Merchants' Shipbuilding Company, Bristol, Pa. A pair of suction and a pair of discharge oil strainers and an oil heater were loaned by the Schutte and Koerting Company. On the discharge pipe of the oil pumps

there was a 6-in. air chamber about eight feet high, which maintained the oil pressure at the burners sufficiently steady for the best results. This chamber was emptied of oil at the beginning of each test and charged with air at 100 lb. pressure.

14 The oil was heated in the railroad-car tank and forced by air pressure into the weighing tank. From this tank the oil after being weighed was discharged into a tank containing a steam coil, and to this tank the pump suctions were connected. Between the

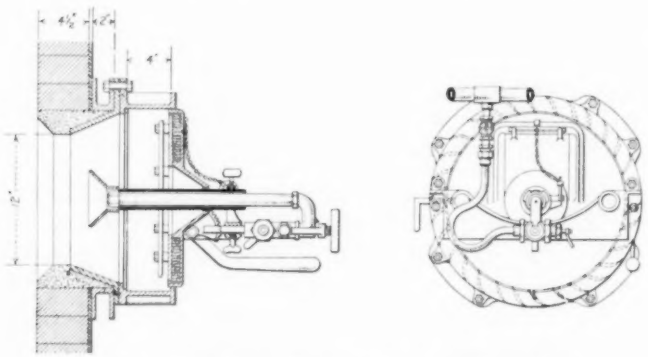


FIG. 4 THE NAVY BUREAU STANDARD BURNER

pumps and suction tank the suction strainer was located and the discharge strainer was placed between the pumps and oil heater.

THE NAVY BUREAU STANDARD BURNER

15 This burner is shown in Fig. 4, and consists of a circular cast-iron register with inclined (not radial) blades around the outside between two thin narrow rings with which they are cast, so formed that the air passes between them somewhat toward the center of the burner, and by them is given rotation.

16 The register is suitably bolted to the boiler front, but between it and the front there is a conical ring which serves to direct the air toward the oil and counteract the opposite tendency caused by the centrifugal effect of the register.

17 The oil is forced through a pipe to the so-called tip, which is screwed on the pipe and serves the purpose of delivering the oil to the furnace in a rapidly rotating stream passing through a very small hole. The tip consists of two hardened steel parts, the outer one being called the "nut", and apparent on the drawing. The

nut has a conical cavity with the apex toward the furnace and the other part is conical and fits it in firm contact. The latter is provided with tangential grooves fed by small holes and these grooves serve to rotate the oil as above mentioned.

18 The centrifugal force derived from the rotation of the oil causes the latter to spread into the form of a thin hollow cone, which is protected from the inrushing air from the register by a cast-iron cone until it reaches an advisable thinness at the edge of the cone, where it is met by the rotating air with which it is thoroughly mixed. Immediately beyond this line of mixture the ignition occurs and the completeness of combustion depends upon the adequacy of the quantity of air, the completeness of the atomization of the oil, and the thoroughness of the mixing.

19 Various adjustments can be made as follows:

a The whole or any part of the register can pass air, or it can be completely closed, by means of a movable circular plate fitting inside of it. This plate can be moved in or out by means of two gear segments and racks operated by a lever

b The burner pipe with tip and spray cone can be moved in or out together

c The spray cone can be moved in or out with respect to the tip.

20 The other burners have tips that rotate the oil, as this is indispensable in thinning the oil film, but burner No. 4 rotates the air but little, and No. 6 not at all. This demonstrates that the rotation of the air is not necessary, a conclusion that might otherwise have been reached.

THE TESTS

21 Four burners of each kind were applied to the boiler, and, except the Navy Bureau Standard burner, were placed with their centers 30 in. apart and 20 in. above the furnace floor, leaving 24 in. between the side walls of the furnace and centers of the adjacent burners. In some of the tests the Bureau burners were arranged likewise, but at the beginning they were placed with their centers 36 in. apart and 22 in. above the furnace floor, leaving 15 in. between the side-burner centers and the walls. No advantage in either spacing could be discerned.

22 The first burner to be used was the Bureau, and on account of inexperience with Mexican oil very poor results were for some

time obtained and immense amounts of smoke made. With more experience excellent results were obtained and there was no difficulty in preventing smoke with this or any burner. In general a slight amount of smoke was allowed.

23 At the beginning there was frequent difficulty from the carbonizing of the oil, but this was gradually overcome.

24 The criteria by which the merits of a burner can be judged when used with induced draft are its ability to burn the oil to CO_2, to do this with a minimum of draft, to produce little smoke, to produce no carbonization in the burner, and to have sufficient range in oil-burning capacity.

25 To produce desirable CO_2 the oil atomization must be well performed and the air admitted in such a manner that it will intimately mix with the oil.

26 A common feature of all the mechanical atomizers tested is the rapid rotation of the oil as it enters the furnace, as in the Bureau burner. This is produced, as in that case, by forcing it at high pressure through small channels of various forms tangential to a small central hole in the burner tip through which it passes to the furnace. As it issues it rotates rapidly and the centrifugal force expands it and forms a thin conical shell of oil. While expanding it is usually protected from the incoming air by a thin cast-iron conical shell of advisable diameter placed in front of it with its base toward the furnace. When the oil reaches the edge of the cone it is swept by the incoming air and the mixing and ignition occur.

CLEANING THE BOILER

27 The boiler had been used for some time in making coal-burning tests. Before beginning the oil tests the outsides of the tubes were thoroughly cleaned by brushing and washing. The insides were not cleaned at first, but before beginning the tests of burner No. 3 the two lower rows of tubes were bored; they were only slightly dirty. Before using burner No. 6 they were all bored out.

STOPPING AIR LEAKS

28 Several times during the tests efforts were made to stop leaks through the casing and between it and the boiler. The CO_2 readings were always higher in the first pass than in the uptake and

TABLE 1 RESULTS OF EVAPORATIVE TESTS OF FOUR-PASS BOILER EQUIPPED WITH FOUR NO. 1 OIL BURNERS

Fuel Oil Used: Mexican Petroleum Residuum

1	Test number	37	33	29	38	31	36
2	Date of trial, 1919	Aug. 8	Aug. 1	Jul. 26	Aug.9	Jul. 30	Aug. 7
3	Duration of trial, hr.	7½	6	8	4.88	7	5
4	Barometer, in.	29.4	29.3	29.5	.....	29.5	29.14
5	Number of burners used	4	4	4	4	4	4
	DIMENSIONS AND PROPORTIONS						
6	Furnace volume, cu. ft.	408	408	408	408	408	408
7	Heating surface, sq. ft.	2518	2518	2518	2518	2518	2518
8	Ratio of h.s. to f.v.	6.13	6.13	6.13	6.13	6.13	6.13
	AVERAGE PRESSURES						
9	Steam pressure by gage, lb.	200.2	197.0	203.1	196.7	193.60	201.0
10	Atmospheric pressure, lb.	14.46	14.26	14.51	.....	14.51	14.34
11	Absolute pressure, lb.	214.66	211.26	217.61	.....	208.11	215.34
12	Draft between damper and boiler, in.	0.382	0.774	0.947	1.475	1.902	2.51
	AVERAGE TEMPERATURES						
13	External air, deg. fahr.	72.8	74.6	82.8	70.6	74.6	86.2
14	Fire room, deg. fahr.	74.7	83.6	89.8	79.2	84.9	92.7
15	Feedwater, deg. fahr.	116	90	68.6	85.9	73.7	91.7
16	Escaping gas, deg. fahr.	414	412	465	516	538	583
17	Furnace by optical pyrometer, deg. fahr.	2198	2230	2331	.....	2551	2654
	FUEL DATA						
18	Oil consumed per hour, lb.	527	615.5	807.8	1137.8	1341.0	1584.6
19	Oil per hour per sq. ft. heating surf., lb.	0.211	0.246	0.325	0.455	0.537	0.634
20	Oil per hour per cu. ft. of fur. vol., lb.	1.292	1.509	1.98	2.79	3.29	3.882
21	Oil per hour per burner, lb.	131.8	153.9	201.95	284	335.3	396.2
22	Temperature of oil at burners, deg. fahr.	266	264.1	276.5	283.4	260	281
23	Pressure of oil at burners, lb.	146.2	181.9	249.3	167.8	228	236.4
24	Heat value per pound of oil, B.t.u.	18376	18376	18352	18376	18376	18376
25	Moisture in oil, per cent	0.15	0.15	0.05	0.15	0.15	0.15
26	Silt, per cent	0.00	0.00	0.00	0.00	0.00	0.00
27	Specific gravity at 60 deg. fahr.	0.962	0.962	0.96	15° C., 0.962	0.962	0.962
28	Viscosity, deg. Engler at 65.5 deg. cent.	20.39	20.39	.....	20.39	.....	20.39
29	Flash temperature, deg. fahr.	167	167	158	167	167	167
30	Burning temperature, deg. fahr.	270	270	266	270	270	270
31	Carbon, per cent	84.02	84.02	84.19	84.02	84.02	84.02
32	Hydrogen, per cent	11.00	11.00	11.11	11.00	11.00	11.00
33	Sulphur, per cent	4.04	4.04	4.07	4.04	4.04	4.04
34	Gravity at 60 deg. fahr., Baumé scale	15.50	15.50	15.80	15.53	15.54	15.54
	QUALITY OF STEAM						
35	Moisture in steam, per cent	1.11	1.04	0.91	0.85	0.87	0.93
	EFFICIENCY						
36	Efficiency of boiler and furnace, per cent	75.9	77.7	76.8	78.9	77.0	76.1
	WATER AND EVAPORATION						
37	Water supplied to boiler per hour, lb.	6663	7783	9876	14520	16555	19620
38	Dry steam generated per hour, lb.	6590	7700	9790	14397	16400	19437
39	Factor of evaporation	1.150	1.176	1.198	1.180	1.192	1.174
40	Evap. per hour from and at 212° fahr., lb.	7580	9060	11730	16988	19550	22819
41	Do. per sq. ft. heating surface, lb.	3.03	3.62	4.69	6.80	7.82	9.14
42	Actual evaporation per lb. of oil, lb.	12.50	12.51	12.12	12.65	12.22	12.27
43	Equivalent from and at 212° fahr., lb.	14.38	14.72	14.53	14.93	14.57	14.40
44	Do. per cu. ft. of furnace volume, lb.	18.57	22.20	24.00	41.63	47.93	55.95
	POWER OF BOILER						
45	Land hp. obtained (rated at 250 hp.)	220	263	340	492	567	661
76	Per cent of land rating obtained	88	104	136	197	227	265
48	Equiv. land hp. of marine rating	435	435	435	435	435	435
44	Per cent of marine rating obtained	51	60	78	113	130	15

TABLE 2 RESULTS OF EVAPORATIVE TESTS OF FOUR-PASS BOILER EQUIPPED WITH FOUR NO. 3 OIL BURNERS

Fuel Oil Used: Mexican Petroleum Residuum

1	Test number	49–A	49	48	50	51
2	Date of trial, 1919	Nov. 6	Nov. 6	Nov. 5	Nov. 7	Nov. 7
3	Duration of trial, hr.	4	2	4.5	3	3
4	Barometer, in.	.....	.....	.....	.....	.....
5	Number of burners used	2	4	4	4	4
	DIMENSIONS AND PROPORTIONS					
6	Furnace volume, cu. ft.	408	408	408	408	408
7	Heating surface, fire sides, sq. ft.	2518	2518	2518	2518	2518
8	Ratio of h.s. to f.v.	6.13	6.13	6.13	6.13	6.13
	AVERAGE PRESSURES					
9	Steam pressure by gage, lb.	200.4	192.4	198.6	198.2	203.0
10	Atmospheric pressure, lb.	14.5	14.5	14.5	14.5	14.5
11	Absolute pressure, lb.	214.9	206.9	213.1	212.7	217.5
12	Draft between damper and boiler, in.	0.286	0.342	0.561	0.925	1.548
	AVERAGE TEMPERATURES					
13	External air, deg. fahr.	43.9	40.7	37.8	42.0	41.7
14	Fire room, deg. fahr.	63.8	61.0	58.3	64.6	66.0
15	Feedwater, deg. fahr.	61.0	60.9	61.0	59	58.1
16	Escaping gases, deg. fahr.	413	474	467	518	562
17	Furnace by optical pyrometer, deg. fahr.	.....	.....	.....	.....	.....
	FUEL DATA					
18	Oil consumed per hour, lb.	522.5	740.5	913.8	1185.3	1435.7
19	Oil per hour per sq. ft. of heating surf., lb.	0.209	0.296	0.365	0.474	0.574
20	Oil per hour per cu. ft. of furnace vol., lb.	1.23	1.82	2.24	2.91	3.52
21	Oil per hour per burner, lb.	261	185	228	296	359
22	Temperature of oil at burners, deg. fahr.	306	322	293	330	325.5
23	Pressure of oil at burners, lb.	210.5	182	198	233.5	248
24	Heat value per pound of oil, B.t.u.	18369	18369	18369	18369	18369
25	Moisture in oil, per cent	0.60	0.60	0.60	0.60	0.60
26	Silt, per cent	0.00	0.00	0.00	0.00	0.00
27	Specific gravity at 15 deg. cent.	0.96	0.96	0.96	0.96	0.96
28	Viscosity, deg. Engler at 65.5 deg. cent.	18	18	18	18	18
29	Flash temperature, deg. fahr.	153	153	153	153	153
30	Burning temperature, deg. fahr.	243	243	243	243	243
31	Carbon, per cent	83.64	83.64	83.64	83.64	83.64
32	Hydrogen, per cent	11.01	11.01	11.01	11.01	11.01
33	Sulphur, per cent	3.68	3.68	3.68	3.68	3.68
34	Gravity at 60 deg. fahr., Baumé scale	15.8	15.8	15.8	15.8	15.8
	QUALITY OF STEAM					
35	Moisture in steam, per cent	1.01	1.07	0.93	0.82	0.80
	EFFICIENCY					
36	Efficiency of boiler and furnace, per cent	76.1	76.1	76.1	75.6	75.6
	WATER AND EVAPORATION					
37	Water supplied to boiler per hour, lb.	6307	8953	11147	14149	17124
38	Dry steam generated per hour, lb.	6243	8857	11040	14033	16897
39	Factor of evaporation	1.206	1.205	1.206	1.209	1.21
40	Evap. per hour from and at 212 deg. fahr., lb.	7529	10673	13314	16965	20554
41	Do. per sq. ft. of heating surface, lb.	3.01	4.27	5.33	6.79	8.22
42	Actual evaporation per lb. of oil, lb.	11.96	11.98	11.94	11.83	11.82
43	Equivalent from and at 212 deg. fahr., lb.	14.41	14.41	14.40	14.31	14.31
44	Do. per cu. ft. of furnace volume, lb.	18.45	26.16	32.63	41.60	50.40
	POWER OF BOILER					
45	Land hp. obtained (rated at 250 hp.)	218	309	386	492	596
46	Per cent of land rating obtained	87	124	154	197	239
47	Equiv. land hp. of marine rating	435	435	435	435	435
48	Per cent of marine rating obtained	50	70	89	113	137

TABLE 3 RESULTS OF EVAPORATIVE TESTS OF FOUR-PASS BOILER EQUIPPED WITH FOUR NO. 4 OIL BURNERS

Fuel Oil Used: Mexican Petroleum Residuum

1	Test number	57	56	58	59
2	Date of trial, 1920	Jan. 15	Jan. 14	Jan. 16	Jan. 16
3	Duration of trial, hr	6	6	6	4
4	Barometer, in	29.29	29.29	29.10	29.10
5	Number of burners used	3–4	4	4	4
	DIMENSIONS AND PROPORTIONS				
6	Furnace volume, cu. ft	408	408	408	408
7	Heating surface, fire sides, sq. ft	2518	2518	2518	2518
8	Ratio of h.s. to f.v	6.13	6.13	6.13	6.13
	AVERAGE PRESSURES				
9	Steam pressure by gage, lb	196.20	195.80	197.60	200.00
10	Atmospheric pressure, lb	14.41	14.41	14.31	14.31
11	Absolute pressure, lb	210.61	210.21	211.91	214.31
12	Draft between damper and boiler, in	0.224	0.547	0.786	1.92
	AVERAGE TEMPERATURES				
13	External air, deg. fahr	17	14.2	.....	.....
14	Fire room, deg. fahr	61	51.2	56	66
15	Feedwater, deg. fahr	42.5	43	42.6	42.0
16	Escaping gases, deg. fahr	427	478	550	632
16	Furnace by optical pyrometer, deg. fahr	.....	.....	.....	.....
	FUEL DATA				
18	Oil consumed per hour, lb	581.8	836.2	1071	1376.3
19	Oil per hour per sq. ft. of heating surf., lb	0.233	0.334	0.428	0.550
20	Oil per hour per cu. ft. of furnace vol., lb	1.43	2.05	2.63	3.37
21	Oil per hour per burner, lb	194	209	268	344
22	Temperature of oil at burners, deg. fahr	324	233.5	319.5	321
23	Pressure of oil at burners, lb	205.3	104.6	208.8	213
24	Heat value per pound of oil, B.t.u	18379	18279	18307	18307
25	Moisture in oil, per cent	0.20	0.40	0.00	0.00
26	Silt, per cent	0.00	0.00	0.00	0.00
27	Specific gravity at 15 deg. cent	0.968	0.962	0.964	0.964
28	Viscosity, deg. Engler at 20 deg. cent	19.2	17.6	19.5	19.5
29	Flash temperature, deg. fahr	151	154	164	164
30	Burning temperature, deg. fahr	257	254	275	275
31	Carbon, per cent	83.80	83.58	83.68	83.68
32	Hydrogen, per cent	11.13	11.16	11.19	11.19
33	Sulphur, per cent	4.00	4.00	4.00	4.00
34	Gravity at 20 deg. cent., Baumé scale	14.6	15.5	15.2	15.20
	QUALITY OF STEAM				
35	Moisture in steam, per cent	0.99	1.03	0.85	0.79
	EFFICIENCY				
36	Efficiency of boiler and furnace, per cent	75.7	75.0	70.8	72.7
	WATER AND EVAPORATION				
37	Water supplied to boiler per hour, lb	6880	9740	11794	15493
38	Dry steam generated per hour, lb	6812	9640	11694	15370
39	Factor of evaporation	1.225	1.224	1.224	1.225
40	Evap. per hour from and at 212 deg. fahr. lb	8345	11800	14313	18823
41	Do. per sq. ft. of heating surface, lb	3.34	4.27	5.72	7.53
42	Actual evaporation per lb. of oil, lb	11.70	11.54	10.90	11.18
43	Equivalent from and at 212 deg. fahr., lb	14.34	14.10	13.36	13.70
44	Do. per cu. ft. of furnace volume, lb	20.47	28.90	35.10	46.15
	POWER OF BOILER				
45	Land hp. obtained (rated at 250 hp.)	242	342	415	546
46	Per cent of land rating obtained	97	137	166	218
47	Equiv. land hp. of marine rating	435	435	435	435
48	Per cent of marine rating obtained	56	79	95	126

TABLE 4 RESULTS OF EVAPORATIVE TESTS OF FOUR-PASS BOILER EQUIPPED WITH FOUR NO. 6 OIL BURNERS

Fuel Oil Used: Mexican Petroleum Residuum

1	Test number	72	71	70	73
2	Date of trial, 1920	Mar. 2	Mar. 2	Mar. 1	Mar. 3
3	Duration of trial, hr.	4	4	5.5	5.5
4	Barometer, in.	29.28	29.28	29.28	29.58
5	Number of burners used	3	4	4	4
	DIMENSIONS AND PROPORTIONS				
6	Furnace volume, cu. ft.	408	408	408	408
7	Heating surface, fire sides, sq. ft.	2518	2518	2518	2518
8	Ratio of h.s. to f.v.	6.13	6.13	6.13	61.3
	AVERAGE PRESSURES				
9	Steam pressure by gage, lb.	196.70	199.40	200.00	198.50
10	Atmospheric pressure, lb.	14.41	14.41	14.41	14.55
11	Absolute pressure, lb.	211.11	213.81	214.41	213.05
12	Draft between damper and boiler, in.	.....	0.368	0.608	1.161
	AVERAGE TEMPERATURES				
13	External air, deg. fahr.	.....	.....	.....	.....
14	Fire room, deg. fahr.	81.4	74.6	72.0	79.4
15	Feedwater, deg. fahr.	44	42	41.4	40.5
16	Escaping gases, deg. fahr.	420	490	522	572
17	Furnace by optical pyrometer, deg. fahr.	.....	.....	.....	.....
	FUEL DATA				
18	Oil consumed per hour, lb.	527	867	1014	1335
19	Oil per hour per sq. ft. of heating surf., lb.	0.211	0.347	0.406	0.534
20	Oil per hour per cu. ft. of furnace vol., lb.	1.291	2.123	2.49	3.27
21	Oil per hour per burner, lb.	176	217	253	334
22	Temperature of oil at burners, deg. fahr.	346.5	340	337	330
23	Pressure of oil at burners, lb.	113	181.7	273.5	229.3
24	Heat value per pound of oil, B.t.u.	18265	18265	18214	18238
25	Moisture in oil, per cent	0.30	0.30	0.50	0.30
26	Silt, per cent	0.00	0.00	0.00	0.00
27	Specific gravity at 60 deg. fahr.	0.961	0.961	0.96	0.96
28	Viscosity, deg. Engler at 65.5 deg. cent.	.....	.....	.....	.....
29	Flash temperature, deg. fahr.	149	149	149	162
30	Burning temperature, deg. fahr.	234	234	237	248
31	Carbon, per cent	83.50	83.50	83.44	83.82
32	Hydrogen, per cent	11.15	11.15	11.23	11.22
33	Sulphur, per cent	3.8	3.8	3.80	3.90
34	Gravity at 60 deg. fahr., Baumé scale	15.7	15.7	15.80	15.80
	QUALITY OF STEAM				
35	Moisture in steam, per cent	1.08	0.95	0.91	0.82
	EFFICIENCY				
36	Efficiency of boiler and furnace, per cent	76.4	76.3	74.8	72.9
	WATER AND EVAPORATION				
37	Water supplied to boiler per hour, lb.	6260	10261	11700	15020
38	Dry steam generated per hour, lb.	6190	10160	11600	14900
39	Factor of evaporation	1.225	1.226	1.226	1.228
40	Evap. per hour from and at 212 deg. fahr.	7580	12460	14240	18300
41	Do. per sq. ft. of heating surface, lb.	3.03	4.99	5.70	7.32
42	Actual evaporation per lb. of oil, lb.	11.75	11.72	11.44	11.16
43	Equivalent do. from and at 212 deg. fahr., lb.	14.39	14.38	14.04	13.70
44	Do. per cu. ft. of furnace volume, lb.	18.57	30.60	34.9	44.90
	POWER OF BOILER				
45	Land hp. obtained (rated at 250 hp.)	220	361	413	530
46	Per cent of land rating obtained	88	144	165	212
47	Equiv. land hp. of marine rating	435	435	435	435
48	Per cent of marine rating obtained	50	83	95	122

it was impossible to prevent it. This is a defect of the water-tube boiler and is not possessed by the Scotch boiler.

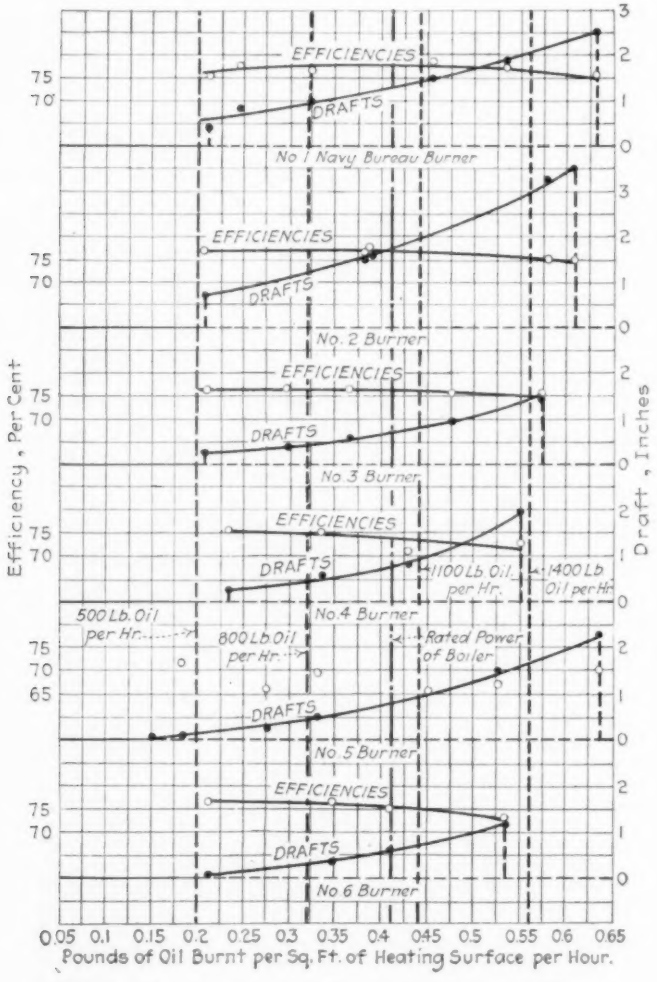


FIG. 5 EFFICIENCIES AND DRAFTS OF SELECTED TESTS

RESULTS OF THE TESTS

29 The results of the tests which are considered to be most reliable and to represent the merits of the burners are given in

Tables 1, 2, 3, and 4. With each burner several preliminary tests were made before sufficient skill was acquired to obtain consistent results, and these are omitted. It may be well to remark here that the representatives of the burners were not able to operate them successfully, which indicates that the efficiencies of boilers with oil fuel are very uncertain.

30 It was intended with each kind of burner to consume oil at four different hourly rates for four burners collectively, namely, 500 lb., 800 lb., 1100 lb., and 1400 lb., but it was impossible to realize these results with precision.

31 In judging the merits of a burner by the magnitudes of the CO_2 and CO, a difficulty is encountered from the fact that these constituents are not uniformly distributed throughout the area in which they are determined, whether it is in the uptake or in one of the passes.

32 Usually, in the boiler tested, the combustion is complete by the time the gases reach the first or lowest pass of the boiler, but if the gases are sampled in this pass they are far from uniform at different points between the end of the baffle and rear header. The following are examples:

Test No.	CO_2 5 in. from header per cent	CO_2 17 in. from header per cent	CO_2 39 in. from header per cent
41	10.6	10.4	11.1
42	11.9	14.2	14.3
43	10.0	12.3	12.6
44	11.9	13.3	13.6
45	11.6		13.6
46	12.6	12.8	12.9

33 While the above are at different distances from the end of the baffle, the following are at the same distance and at different points in a line parallel to the end of the baffle and 19 in. therefrom:

Test No.	Position C CO_2 per cent	Position D CO_2 per cent	Position E CO_2 per cent
26	11	12	12.3
36	12.9	9.5	11.8

Similarly the other ingredients of the gases vary.

34 This variation of the gases of combustion renders it difficult to apply the chemical criterion in determining merits of a burner. In operating burners, of course, constant studies of the gas analyses

are made, and this is necessary, for if the CO_2 is good at one point it is probable that it is at others. Nevertheless, it is necessary to determine the efficiency of the boiler under the same conditions as a final means of judging of the performance of a burner. It may

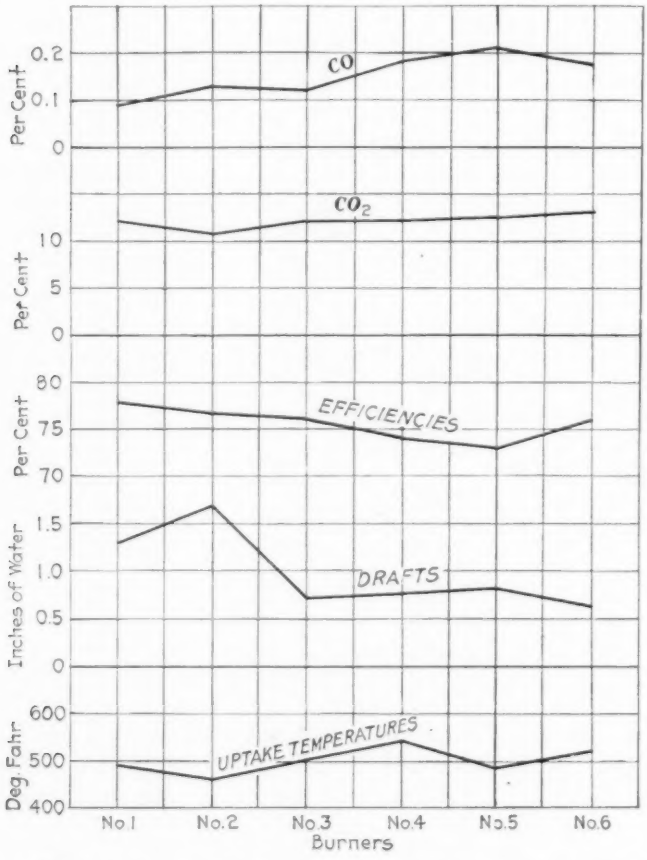


FIG. 6 COMPARISON OF BOILER EFFICIENCIES, DRAFTS, CO_2 AND UPTAKE TEMPERATURES AT RATED POWER OF BOILER WITH DIFFERENT BURNERS, AND AVERAGE CO OF SELECTED TESTS

be found that the maximum boiler efficiency will not correspond with the maximum CO_2, and this is true with these tests.

35 In the tables of results given the following efficiencies and average CO_2 and CO occur, and are here listed in the order of the burner trials.

Burner	Average efficiencies, per cent	Average CO_2 per cent	Average CO per cent
No. 1	77.02	11.88	0.09
No. 3	75.94	12.16	1.12
No. 4	73.50	12.00	1.18
No. 6	75.10	12.71	0.18

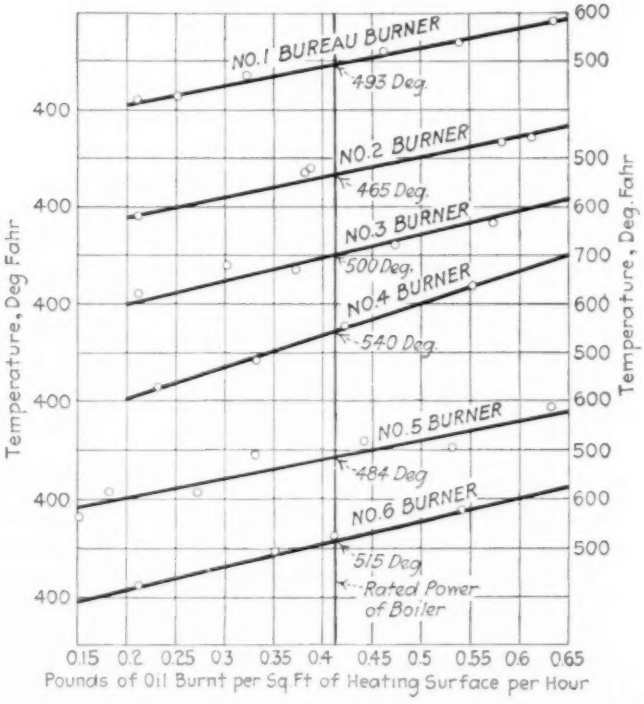


FIG. 7 TEMPERATURE GRADIENTS FOR DIFFERENT RATES OF OIL CONSUMPTION WITH DIFFERENT BURNERS

36 In Fig. 5 the efficiencies and drafts of the various burners are plotted. The Bureau Standard burner required more draft than the others. The reason for this is that it was designed for use on war vessels and made to require a strong draft, or fire-room pressure, in order not to backfire from gun shock.

37 In Fig. 6 a diagram of comparative uptake temperatures, drafts, efficiencies, CO_2 and CO is given.

38 In Fig. 7 the uptake temperature gradients of the different

burners are shown. It will be noticed that the burner of lowest efficiency gave the steepest gradient. In Figs. 5, 6 and 7 all burners are included.

39 Burner No. 6 required the least draft and the least adjusting. It was the only burner that did not vibrate; it gave the highest CO_2, but also next to the highest CO. All other burners required constant adjusting to maintain proper CO_2, and the reason for this could not be ascertained.

DISCUSSION

C. B. Goode. The author speaks of trouble with the brickwork, but did not go into any details; the temperatures, between 2200 and 2600 deg. fahr., which he quotes, hardly give enough information to determine the trouble. The writer has experienced considerable trouble with brickwork in oil-burning furnaces, especially with the pressure burners, and in larger furnaces. The larger the furnace, the more difficult it is to keep the brickwork intact, and with a small firebox like this, it does not seem as though there would be much trouble with temperatures between 2200 and 2600 deg. fahr.

Joseph Nelis. The writer was connected with Mr. Dean in the early part of these tests. An attempt was made with this standard-size furnace to run up to as high a capacity as possible. At some of the higher capacities the brickwork melted. In connection with this set of tests it must be remembered that in marine work there is a fixed load condition. This was a marine boiler and was put in the plant in Erie, Pa., and tested for its efficiency at various ratings. The boiler was operated at higher ratings than it would be operated on a ship. At the lower ratings there were no brickwork troubles.

Another object was to test different burners as well as the boilers. It should be kept in mind that these oil burners and boiler were designed for a fixed load condition, and that it is necessary on a ship to have as small a furnace as possible, because there is only a certain amount of headroom for the boiler and furnace.

The burners which are used on all modern ships are necessarily mechanical atomizing burners. We cannot use steam atomizers, because fresh water is important and that forces us to use the mechanical atomizing burners. Up to the ratings which are used in

marine work, about 4.5 to 6 lb. evaporation per square foot of heating surface, with proper temperature and pressure of oil it is possible to operate without smoke.

H. E. Lewis. The writer has made a comparison of the tables given in the paper with some figures obtained on a 450-hp. water-tube boiler with 4-in. tubes 14 ft. long. The height of furnace at the front was 7 ft. 4 in.; at back, 5 ft. 7 in.; width of furnace, 9 ft. 10 in.; distance between front and rear walls, 7 ft.; furnace volume, 445 cu. ft.; furnace volume per burner, 148 cu. ft. The efficiencies at various percentages of rating were as follows:

Per cent of rating	116.4	156.3	199.9	251.8	321.7	372.0
Efficiency, per cent	81.8	81.4	79.05	77.9	75.29	71.58

These burners used 15-deg. B. oil, the oil per burner per hour ranging from 390 to 1439 lb., or 0.98 cu. ft. per rated boiler horsepower. The evaporation ranged from 40.6 lb. to 129.8 lb. water from and at 212 deg. per cu. ft. of furnace volume.

With a 3-in. tube marine-type boiler the following efficiencies at land ratings were obtained:

Per cent of rating	104	105	155	212	274	355
Efficiency, per cent	82.05	81.30	79.70	79.35	77.00	76.50

This boiler had 2755 sq. ft. heating surface and on land rating would be 275.5 horsepower. The furnace volume was 239 cu. ft. with these burners or 79 cu. ft. per burner. At the above ratings, 41.3 lb. of water from and at 212 to 140.7 lb. per cu. ft. of furnace volume was obtained at good efficiencies.

There is a statement in the latter part of Par. 30 which indicates that the efficiency of boilers with oil fuel are very uncertain. The writer believes that a well-designed mechanical oil burner will give higher efficiencies with less care and manipulation on the part of the operator than any other fuel-burning apparatus. The adjustment of air and oil pressure is practically all that a man has to watch, other than seeing that the burners are free.

John A. Stevens. When the oil installation was first put on the *Mauretania*, before periscopes were provided to observe the

smoke, there was an understanding between the deck officers and engineers whereby the engine room was informed of the amount of smoke issuing from the funnels.

When the *Mauretania* was laid up last winter, periscopes were installed in the uptakes, consisting of a high-powered nitrogen lamp and a piece of plate glass on one side and a mirror at 45 degrees on the other side of the uptakes. This arrangement would inform the engineers of the character of the smoke passing up by the periscope. In this way the engineers could regulate the amount of smoke. It incidentally made quite a saving in the oil burned.

Instead of pails in front of the boilers, there were open boxes 3 or 4 in. wide and 3 ft. long filled with sand, which caught the dripping oil.

The ship is equipped with a system of foamite tanks and CO_2 recorders. A particularly noticeable feature is the cleanliness of the fireroom.

No. 1811

THE VERTICAL TRIPLE-EXPANSION PUMPING ENGINE

A NEW RECORD PERFORMANCE AT CLEVELAND, OHIO

By L. A. Quayle, Cleveland, Ohio

and

E. H. Brown, Milwaukee, Wis.

Members of the Society

Cleveland, Ohio's, installations during the past sixty-five years make an interesting study of both the improvement which has been made in the economy of pumping engines, and the increase in energy available to the engine which has resulted from successive increases in steam pressure and temperature. These installations also cover the period of the introduction of stoker firing and the operating records show their effect on the overall economy.

In 1917 Cleveland's new Division Avenue Pumping Station was put into service. The results of recent tests of the units of this station are given in detail in the present paper, as well as results of the official tests of all of the nine other vertical triple-expansion pumping engines in service in the Cleveland Water Works.

The authors' investigation leads them to believe that the Division Avenue station contains at least one and possibly four engines whose duty record and thermal efficiency have never been equalled by any other steam-driven pumping engine, namely, 211,000,000 ft-lb. of work per 1000 lb. of steam used and 24.3 per cent thermal efficiency. They consider that the development which has led to such high efficiency in water-works steam pumping engines that require in general a maximum of not over 1500 hp., has been largely the result of the practice, instituted by water-works engineers, of making economy tests on all installations.

THE crank-and-flywheel vertical triple-expansion pumping engine originated from the designs of high-duty pumping machinery made under the direction of the late Edwin Reynolds. The first engine of this type was installed in the Milwaukee Water Works in 1886, is still in constant service, and differs but little in appearance and general design from the most modern unit. The high

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mechanical and thermal efficiencies of Reynolds' design were early recognized, and many elaborate and accurate tests were made on different installations. The first which attracted general attention were made by R. C. Carpenter and published in the Transactions of the Society in 1893.[1] The Reynolds design has been adopted by water-works engineers as a standard for American practice, and has been extensively copied abroad.

2 The introduction and adoption for general water-works service of this slow-piston-speed type of unit with its resulting enormous weight per unit of power, at a time when the whole tendency of prime-mover development was toward high speed and moderate unit weight, as evidenced by the extensive use of the Holly Gaskill type of pumping engine, the Porter-Allen high-speed general-service engine, and also the introduction of the Parsons turbine for small-generator drive, would have been impossible were it not for the very high efficiency over wide variations of capacity and head, and the great reliability inherent in pumping units of this type.

3 So far as the authors know, there is no record of any triple-expansion pumping engine having been wrecked or broken beyond standard commercial repair, and there are records available of engines of this type having run continuously with only a few hours' shutdown for adjustment and minor repairs for a period of over ten years.

4 Cleveland, Ohio's, installations during the past 65 years make an interesting study of both the improvement which has been made in the economy of pumping engines, and the increase in energy available to the engine which has resulted from successive increases in steam pressure and temperature. These installations also cover the period of the introduction of stoker firing and the operating records show their effect on the overall economy. The equipment which has been used and the station duties obtained during these years (1856 to 1921) is listed in Table 1.

5 It will be observed from this table that the duty of 53,000,000 ft-lb. per 100 lb. of coal obtained by the very-slow-speed but well-made Cornish engines with which the Water Works began operation in 1856, using 22 lb. gage steam pressure, was better than was obtained from the Knowles engines purchased 31 years later and operated at a boiler pressure over three times greater.

6 The first decided increases in Cleveland's pumping-engine duties came with the installation in 1893 of the Worthington high-duty engine, a type in which the steam was worked expansively in

[1] Trans., Am. Soc. M.E., vol. 14, p. 426.

a multiple-cylinder compound engine without a crankshaft and flywheel, and with the installation of the Allis, Holly, and Kilby vertical triple-expansion crank-and-flywheel engines in 1898 to 1901.

7 A study of Table 1 is also of interest from the standpoint of engine and boiler depreciation. The engine longest in service was Cornish No. 2, which was put in operation at Division Avenue in 1856, moved to Fairmount Station in 1885, and taken out of

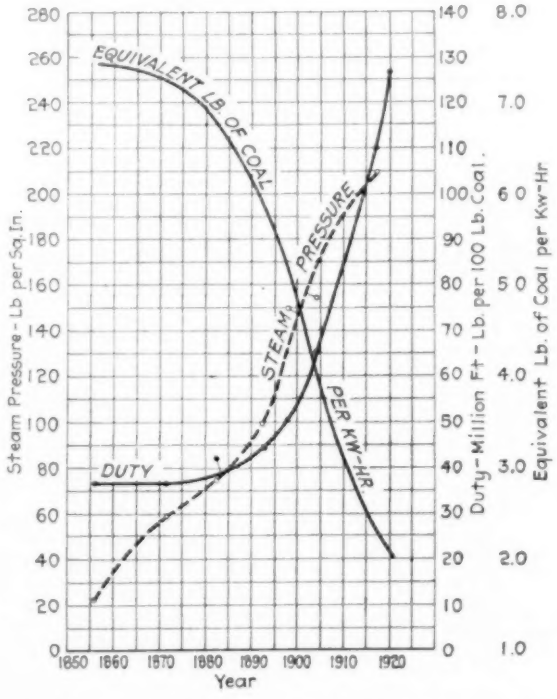


FIG. 1 STEAM PRESSURE AND ECONOMY, CLEVELAND'S PUMPING STATIONS, 1856–1921, NO ECONOMIZERS

service in 1905, making a total of 49 years from the date of first operation until it was permanently shut down because of obsolescence. Worthington engine No. 3 was installed in 1884 and is still in service at Fairmount Station, a total of 37 years to date. The boilers longest in the Water Department service were the Cornish purchased for Division Avenue in 1872, moved to Fairmount Station in 1894, and finally dismantled in 1905, making a total of 33 years from the time

TABLE 1 EQUIPMENT AND DUTIES OF CLEVELAND PUMPING STATIONS SINCE 1856.

	Engines				Boilers						Stokers		Duty
Name of Pump	Type	Capacity in Million Gallons per Day	Put in Service	Dismantled Moved, Rebuilt	Station Boiler Pressure and Superheat	Duty of Engine and Boiler, Million ft-lb.	Type Boiler	No. and Size of Boiler	Date Installed	Dismantled Moved	Date Installed	No. Installed and Name	Yearly Average Station Duty per 100 lb. Coal as Fired, Million ft-lb.
Division Avenue Station — 1856–1872 — 17 Years' Operation													
Cornish No. 1 and No. 2	Walking beam.....	4	1856	No. 1-D-1885 " 2-M-1885 D-1905	22	(c) 53	Cornish	6 6′−0″ × 28′−0″	1856	D-1882		Hand-fired	37
Division Avenue Station — 1872–1905 — Low-Duty Section													
Cuyahoga No. 1.....	Horiz. comp. duplex	10	1872	D-1893	60	(c) 42	Cornish	6 6′−0″ × 28′−0″	1872	M-1884 D-1905		Hand fired	
Worthington No. 1.....	Horiz. comp. duplex	10	1876	D-1900		(c) 54	H. R. T. Marine Marine	1 4 3	1876 1878 1880	D-1895 D-1895 D-1896		Hand-fired	
Worthington No. 2.....	Horiz. comp. duplex	10	1882	D-1905	75	(c) 60	Marine	3	1883	D-1897		Hand-fired	40
Worthington No. 3....	Horiz. comp. duplex	10	1884	M-1905		(c) 56						Hand-fired	
Knowles Nos. 1 & 2....	Horiz. comp. duplex	15	1887	M-1905 D-1911		(c) 48	H. R. T. H. R. T.	6 and 3 8′−0″ × 16′−0″	1887 1892	D-1905	1893	3 Murphy	

Division Avenue Station — 1893-1905 — High-Duty Section													
Worthington No. 4	Horiz. comp. duplex high-duty	15	1893	M-1905 D-1917	100	(c) 76	H. R. T.	6 6′−6″ × 16′−0″	1893		1893	6 Murphy	65
							H. R. T.	13 6′−4″ × 18′−0″	1895	M-1905 D-1912	1895	8-American 5-Brightman	
Allis No. 1	V. T. E. C. & F.	20	1898	R-1915	150	86 (c)	Geary	6 220 b.hp	1898	D-1914	1898	6-Brightman	
Holly No. 1	V. T. E. C. & F.	10	1901	R-1915		95 (c)							
Kilby No. 1	V. T. E. C. & F.	20	1902	R-1915		86 (c)	Stirling	2 450 b.hp	1902	D-1914	1902	4-American	
Kirtland Pumping Station — 1904 to Date													
Holly Nos. 2 and 3	V. T. E. C. & F.	25	1904	In service		118 (c)	B. & W. Stand	8 272 hp.	1904	In service	1904	S-B. & W. Chain grate	96
Holly Nos. 6 and 7	V. T. E. C. & F.	25	1912	In service	155 lb 120°	125 (c)							
Delaval No. 1	Turb'n. geared cent	30	1914	(a)									
Division Avenue Station — 1917 to Date													
Allis No.1	V. T. E. C. & F.	20	1917	R-1915	210 lb. 120°		Stirling	6 500 b.hp.	1916	In service	1916	6-Riley Underfeed 6-Retort	(b) 127
Holly No. 1	V. T. E. C. & F.	10	1917	R-1915									
Kilby No. 1	V. T. E. C. & F.	20	1917	R-1915									
Allis-Chalmers No. 2	V. T. E. C. & F.	25	1917	In service									
Allis-Chalmers No. 3	V. T. E. C. & F.	25	1917	In service									
Allis-Chalmers No. 4	V. T. E. C. & F.	20	1917	In service									

(a) Has operated 650 hr. since New Division Ave. Station was put in operation in 1917. Allis No. 1, Holly No. 1, Kilby No. 1, rebuilt in 1915 for higher steam pressure.

(b) Duty of turbine-driven centrifugal pumps in low-lift room not included.

(c) Per 100 lb. dry coal over 24-hr. period.

of purchase until they were dismantled with the Cornish engines because of obsolescence.

8 This table of installations also shows that the steam pressure which for the first installation in 1856 was 22 lb. gage, was raised to 60 lb. in 1872; to 75 lb. in 1883; to 100 lb. in 1893; to 150 lb. in 1898, and to 210 lb. when Division Station was rebuilt and put into service in 1917.

9 In 1892 at Division Avenue Station all boilers were hand-fired, 17 of them being in service and requiring 42 firemen for a 24-hour day. The station started using overfeed stokers of the

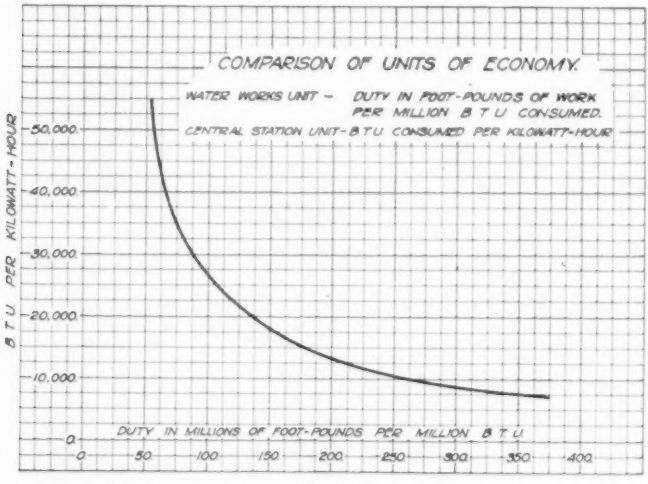


FIG. 2 COMPARISON OF UNITS OF ECONOMY

Water-works unit, duty in foot pounds of work per million B. t. u. consumed; central-station unit, B. t. u. consumed per kw-hr.)

Murphy type in 1892, purchased underfeed stokers of the American type in 1895 and 1896, overfeed stokers of the Brightman type in 1897 and 1898, and went back to underfeed stokers in 1902. Chain-grate stokers were installed in the new Kirtland Station in 1904, but the department again returned to the use of underfeed stokers in 1915 by their purchase for the new Division Avenue Station.

10 The effect of increased steam pressure, improvement in boiler and stoker efficiencies, and the changes in pumping-engine design are recorded in the duty columns of Table 1 and have been summarized graphically in Fig. 1, which shows the steam pressure

together with the duty obtained from the different installations, plotted as a function of the time in years. It is interesting to observe that the increase in steam pressure from 22 lb. in 1856 to 80 lb. in 1883, did not result in higher duties owing to the fact that Cleveland purchased duplex pumps with the first increase in steam pressure. The purchase of this type of pump probably resulted from the

TABLE 2 COMPARISON OF ECONOMY RECORDS AS GIVEN IN KENT'S HANDBOOK WITH DIVISION AVENUE STATION SPECIFICATIONS ADDED

Date of test.	1899	1900	1900	1901	1906	1915	1918	1918
Locality....	Wildwood, Pa.	St. Louis, Mo.	Boston, Chestnut Hill	Boston, Spot Pond	St. Louis, Bissell's Point	Cleveland, Division Ave.	Cleveland, Division Ave.	Cleveland, Division Ave.
Expansion...	Quadruple	Triple	Triple	Triple	Triple	Triple	Triple	Triple
Figures based on..	Test	Test	Test	Test	Test	Specifications	Test under specifications conditions	Test using receiver heater
Ft-lb. of work done per million B.t.u. consumed....	162.9	158.07	156.8	156.59	158.85	170	176.3	189.2
Corresponding thermal efficiency, per cent [1]..	20.95	20.32	20.15	20.13	20.42	21.95	22.61	24.27
Equivalent B.t.u per kw-hr.,[2] including all condenser and other auxiliaries.	16280	16780	16930	16940	16700	15600	15070	14030

[1] The thermal efficiency in water-works practice is computed from an actual mean condensate temperature and not on the assumption that the condensate leaves the engine at the temperature of the exhaust steam, and also includes the power required for all the engine's auxiliaries.

[2] For the relation between foot-pounds of work per million B.t.u., and B.t.u. per kw-hr. see curves in Fig. 2.

low prices due to the adoption of the duplex direct-acting Worthington design by a large number of pump builders following the expiration of the basic patents.

11 It is interesting to note in connection with boiler-room obsolescence that at the Division Avenue Station in 49 years of active service (1856 to 1905) there were 14 different boiler installations, making a total of 53 boilers of six different types. This station

started out in 1856 with the same number of boilers (six) for a pumping-station capacity of 8,000,000 gal. per day as the new station is now provided with for a capacity of 150,000,000 gal. per day.

12 In 1904 Kirtland Pumping Station was put in service using 155 lb. gage boiler pressure and 120 deg. superheat, and with a greatly improved boiler plant which resulted in an increased station duty per 100 lb. of coal of approximately 47 per cent as compared with the best Division Avenue Pumping Station records, although the engine duties per 1000 lb. of steam were not greatly increased, as is shown by the official duty tests listed as Item No. 71 in columns Nos. 10, 11, 14, 15, and 16 of Table 3. Eight years of progress in pumping-engine design are reflected in the increase in duty of Holly engines Nos. 6 and 7 of about 9 per cent when tested under approximately the same pressure and superheat as Holly engines Nos. 2 and 3, a portion of the gain resulting from a better vacuum due to the use of water-works-type surface condensers instead of the jet type with which Holly engines Nos. 2 and 3 were equipped.

13 In 1917 Cleveland's new Division Avenue Pumping Station was put into service. As far as the authors have been able to determine, the average duty of this station with a performance equivalent to 127,000,000 ft-lb. of work per 100 lb. of coal fired is not exceeded by any other coal-fired pumping plant in which the normal low cost of coal does not warrant investment in economizers. This low coal consumption is due to the use of efficient boiler and stoker equipment operated under constant engineering supervision and the use of highly efficient triple-expansion engines working under 210 lb. steam pressure and 120 to 150 deg. superheat.

14 Our investigation also leads us to believe that this station contains at least one and possibly four engines whose duty record and thermal efficiency have never been equaled by any other steam-driven pumping engine. The best economy obtained on any one of these four engines — 211,000,000 ft-lb. of work per 1000 lb. of steam used and 24.3 per cent thermal efficiency — is compared in Table 2 with a summary published in Kent [1] of the most notable high-duty pumping-engine records.

15 It is instructive to compare the thermal efficiency (including auxiliaries) of 24.3 per cent obtained on the 1400-hp. vertical triple-expansion pumping engine with 24.8 per cent, the maximum thermal efficiency (excluding auxiliaries) of the Interborough Rapid Transit 45,000-hp. cross-compound turbo-generator; with 24.1 per cent,

[1] Mechanical Engineers' Pocket-Book, 9th Edition, p. 806.

the maximum thermal efficiency obtained by any one of four 40,000,-000-gal.-per-day low-head 300-hp. Humphrey gas pumps at Chingsford, England; and also with the 24.1 per cent thermal efficiency obtained on a 1500-hp. natural-gas-engine generating unit of the

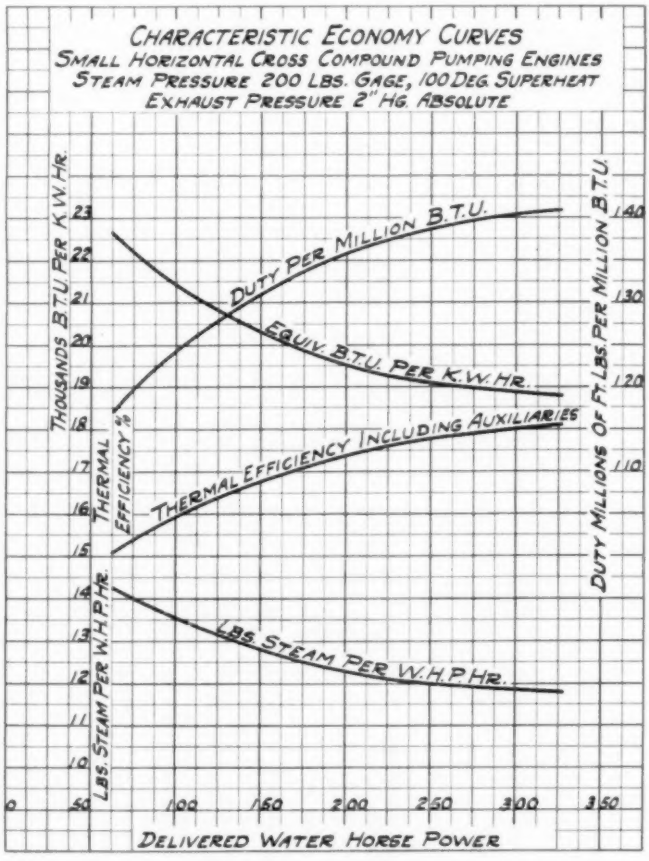


FIG. 3 CHARACTERISTIC ECONOMY CURVES OF SMALL HORIZONTAL CROSS-COMPOUND PUMPING ENGINES

(Steam pressure, 200 lb. gage; superheat, 100 deg., exhaust pressure, 2 in. Hg abs.)

American Locomotive Co. at Allegheny, Pa., as it shows that these four distinctly different types of units have all reached a point of development at about the same time at which they give practically the same thermal efficiency. This would seem to indicate that any

TABLE 3 DIMENSIONS OF CLEVELAND'S 12 V.T.E. PUMPING ENGINES AND RESULTS OF 16 OFFICIAL TESTS

Engine	A.C. No. 4	A.C. No. 4	A.C. No. 3	A.C. No. 2	Allis No. 1	Kilby No. 1	Holly No. 1	Holly No. 6	Holly No. 7	Holly No. 2	Holly No. 3	Holly No. 4	Holly No. 5	Kilby No. 1	Holly No. 1	Allis No. 1
Station and Year Installed or Rebuilt	Division 1917		Division 1917	Division 1917	Division 1916	Division 1916	Division 1916	Kirtland 1912	Kirtland 1912	Kirtland 1903	Kirtland 1903	Fairmount 1907	Fairmount 1907	Old Division 1902	Old Division 1902	Old Division 1898
1. Number of steam cylinders	3		3		3	3	3	3		3		3				
2. Diam. of steam cylinders, in.	36–68–108		32–60–96		29.5–58–92	28–54–83	27-52-84	34–64–98		32–60–90		12.75-24-36		30–54–83	30–56.5–84	34–62–92
3. Diam. of piston rods, in.	7½–7½–8½		6¾–6¾–8½		6¾–7–7	7¼–7–7	7½–7–7	8		7.5		2.75		7	7	4¼–4¼
4. Stroke of steam pistons and water plungers, in.	66		66		64	64	48	66		60		24				
5. Net area of h.-p. steam piston, sq. in.	976.6		764.5		651.6	595.1	551.9	882.8		782.2		124.7		687.6	687.6	891.1
6. Net area of i.-p. steam piston, sq. in.	3590		2792		2604	2271	2104	3192		2805		449.4		2271	2488	3002
7. Net area of l.-p. steam piston, sq. in.	9133		7206		6608	5391	5523	7518		6340		1015		5392	5523	6630
8. Proportion of net area of pistons, ratio	1:3.68:9.35		1:3.63:9.38		1:40:10.15	1:3.81:9.06	1:3.81:10.0	1:3.616:8.52		1:3.60:8.11		1:3.60:8.14		1:3.30:7.84	1:362:8.03	1:3.37:7.44
9. No. of water plungers (single-acting)	3		3		3	3	3	3		3		3				
10. Diameter of each plunger, in.	31.63		34.75		34	33	22.88	34.75		36		9.875				
11. Area of each plunger, sq. in.	785.5		948.4		907.9	855.3	411.0	948.4		1018		76.59				
12. Displacement of one plunger per revolution, 1000 cu. in.	51.84		62.60		58.11	54.74	19.73	62.60		61.0		1.838				
13. Displacement of three plungers per revolution, 1000 cu. in.	155.5		187.8		174.3	164.2	59.19	187.8		183.1		5.514				
14. Displacement of three plungers per revolution, cu. ft.	90.01		108.7		100.9	95.03	34.25	108.7		106.0		3.19				
15. Displacement of three plungers per revolution, gal.	673.3		812.9		754.6	710.9	256.2	812.9		793.1		23.87		Same as 1916	Same as 1916	Same as 1916
16. No of valve chambers	6		6		6	6	6	6		6		6				
17. Diameter of each valve chamber, in.	47.25		47.25		63.5	63	47	59.75		59						
18. No. of valves in each cage and no. of cages per deck	28	7	35	7	35	35	15	15	6	15	6	37				
19. Diam. of valve openings, in.	3.25		3.25		3.0	3.0	3.75	3.75		3.75		2.75				
20. Area of valve openings, sq. in.	8.295		8.295		7.068	7.068	11.04	11.04		11.04		3.97				

21. Net area of waterway through set of valves, sq. ft.	8.166	10.21	7.656	7.656	3.39	10.0	10.0	1.020			
22. Ratio of net valve area to plunger area, ratio	1.497	1.55	1.212	1.289	1.189	1.518	1.415	1.918			
23. Diam. of air pump, in.	30	30	28	28	24	24	28	8			
24. Stroke of air pump, in.	66	66	64	48	48	66	60	24			
25. Type of condensers	Surface	Surface	Surf.	Surf.	Surf.	Surface	Jet	Surface	Jet	Jet	Jet
26. No. of condensers	1	1	1	1	1	2	1	1			
27. Diam. of condenser (inside), in.	62	62	62	62	53	60	.....	30.13			
28. No. of tubes in each condenser	790	790	790	790	572	603	.....	135			
29. Inside diam. of tubes in condenser, in.	1	1	1	1	1	0.87	.....	1 (O. D.)			
30. Length of tubes in condensers (between heads), ft.	10.5	8.667	6.959	6.959	7.25	6.417	.....	7.292			
31. Tube surface in condenser, sq. ft.	2200	1800	1450	1450	1100	1762	.....	250			
32. Diam. of main shaft at bearing, in.	20	20	18	18	16	20	17.5	7.5			
33. Diam. of main shaft at flywheels, in.	22.5	22.5	20.25	20.5	19	23.31	20.75	9			
34. Number of flywheels	2	2	2	2	2	2	2	2			
35. Diameter of flywheels, ft.	20	20	20	20	16	20	20	8.5			
36. Diameter of suction pipe, in.	42	48	36	36	30	48	48	14			
37. Diameter of discharge pipe, in.	42	42	36	36	30	42	42	12			
38. Weight of one flywheel, tons	37.5	32.5	32.35	.....	24	38	31.5	3.4			
39. Total weight of engine, tons	950	875	.....	.....	.....	970	970	110			
40. R.p.m. when pumping at rated capacity	20.61	21.35	18.45	19.53	27.10	21.36	21.89	58.18			
41. Piston speed at rated capacity, ft. per min.,	226.7	234.9	196.8	208.2	216.8	234.9	218.9	232.7			
42. Piston speed at rated capacity, ft. per sec.,	3.778	3.91	3.279	3.47	3.61	3.915	3.65	3.879			
43. Velocity of flow per second, through suction and discharge valves, ft.	1.267	1.262	1.347	1.347	1.52	2.58	2.58	2.022			
44. Velocity of flow ft. per sec. through suction pipe,	3.216	3.077	4.374	4.374	3.15	3.08	3.08	2.895			
45. Velocity of flow ft. per sec. through discharge pipe,	3.216	4.018	4.374	4.374	3.15	4.02	4.02	3.940			

TABLE 3 (*Continued*) — TEST DATA

Engine	A.C. No. 4	A.C. No. 4	A.C. No. 3	A.C. No. 2	Allis No. 1	Kilby No. 1	Holly No. 1	Holly No. 6	Holly No. 7	Holly No. 2	Holly No. 3	Holly No. 4	Holly No. 5	Kilby No. 1	Holly No. 1	Allis No. 1
Column	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
Date of test	10–14 1918	10–15 1918	10–5 1918	10–10 1918	11–7 1918	11–21 1918	11–15 1918	11–4 1912	11–13 1912	7-20&21 1904	7-13&14 1904	12–13 1907	12–5 1907	12–14 1902	11–7 1902	3–14 1899
Condensate heaters in service	Exh.	Exh., Rec.														
Duration of test, hr.	8	8	8	8	2	2	4	12	12	24	24	12	12	24	24	24.04
Average Pressure and Temperatures																
1. Steam pressure at throttle, lb. gage	206.3	207.1	203.8	208.4	195.8	200.2	199.3	162.9	164.8	149.1	146.8	152.3	149.7	153.9	152.0	149.4
2. Barometric pressure at 70 deg. fahr., in. Hg.	29.61	29.73	29.28	29.59	29.7	29.50	29.6	29.51	28.89	29.24	29.25	28.75	28.95	29.62	29.3	29.68
3. Barometric pressure (corrected to 32 deg. fahr.), lb.	14.48	14.54	14.32	14.47	14.51	14.43	14.48	14.49	14.18	14.38	14.39	14.13	14.20	14.49	14.35	14.52
4. Steam pressure at throttle, lb. abs.	220.8	221.7	218.1	222.8	210.3	214.6	213.8	177.4	179.0	163.5	161.2	166.4	163.9	168.3	166.4	164.0
5. Steam pressure in first receiver, lb. gage	33.7	35.8	34.6	34.0	35.5	38.0	33.3	27.64	27.85	28.24	29.35	26.14	26.13	33.70	25.88	23.86
6. Steam pressure in second receiver, lb. abs.	13.2	11.7	15.8	15.8	13.53	12.71	12.02	10.49	10.34	11.04	10.85	10.95	10.54	14.88	20.05	10.45
7. Vacuum in condenser referred to 30-inch barometer, in. Hg	28.20	28.26	28.43	28.66	28.42	28.67	28.04	27.69	27.27	24.50	24.79	27.10	26.45	26.59	26.44	27.74
8. Temperature of steam at throttle, deg. fahr.	520.6	526.6	503.5	495.6	506.8	499.7	490	493.6	493.6	472.1	464.1	366.7	365.5	367.6	366.7	365.0
9. Degrees superheat at throttle, deg. fahr.	130.4	136.0	114.4	104.6	120.7	111.8	102.5	118.7	117.9	104.6	97.66	none	none	none	none	none
10. Temperature of exhaust steam, deg. fahr.	97.9	94.8	92.6	89.4	93.7	89.0	99.2	105.5	101.4	134.2	135.3	116.4	124	120	121	105
11. Temp. of main condensate entering exh. heater, deg. fahr.	67.4	72.9	70.0	68.9	62.5	60	60	.....	.....	.....	.....	70.73	88	.....	.....	.....
12. Temp. of main condensate leaving exh. heater, deg fahr.	94.8	92.8	89.3	86.9	90.4	85.3	96.5	.....	.....	.....	.....	103.9	112.6	.....	.....	.....
13. Temp. of main condensate leaving receiver heater, deg. fahr.	.....	199.2	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
14. Rise of temp. through exhaust heater, deg. fahr.	27.4	19.9	19.31	17.97	27.91	25.3	36.5	.....	.....	.....	.....	33.14	24.6	.....	.....	.....
15. Rise of temp. through receiver heater, deg. fahr.	.....	106.3	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
16. Temp. of jacket and drain water, deg. fahr.	207.9	161.8	210.3	210.9	190.7	211.0	189.9	164.5	161.0	190.2	184.9	189.1	186.8	222.2	183.7	.....
17. Temp. of water pumped, deg. fahr.	61.5	61.5	61.0	60.0	54	53	53	55	54	73.8	69.5	41.8	42.6	39.6	58	34
18. Water pressure in force main, lb.	167.0	169.9	115.5	114.6	110.7	106.6	168.2	96.89	97.48	73.38	71.46	183.0	180.7	76.87	170.4	85.34
19. Water pressure in suction main, lb.	2.29	3.86	3.42	3.79	6.58	5.78	5.03	.....	.....	.....	.....	6.0	5.63	.....	.....	.....
20. Elev. of suct. gage above disch. gage, ft.	3.09	3.95	3.40	4.32	−7.58	−4.5	−6.25	.....	.....	.....	.....	.....	.....	.....	.....	.....

21. Disch. head above center line	388.7	393.2	269.5	267.5	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
22. Suction head above center line of valve decks, ft.	11.54	16.03	14.14	15.89	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
23. Total head pumped against, ft.	377.2	377.2	255.4	251.6	248.0	237.2	382.9	250.5	251.6	194.2	188.9	408.3	404.0	199.1	405.3	202.3
Total Quantities																
24. Total main condensate, 1000 lb.	91.94	82.87	78.06	77.66	14.96	14.29	24.28	123.4	127.5	218.9	210.6	19.76	19.21	190.8	194.9	.....
25. Total jacket and drain water, 1000 lb.	9.060	17.91	8.840	9.097	1.758	1.613	2.151	12.27	11.71	22.04	22.84	3.189	3.074	27.07	23.31	.....
26. Total dry steam used by unit, 1000 lb.	101.0	100.8	86.90	86.76	16.72	15.91	26.43	135.7	139.2	238.0	229.7	21.36	22.58	217.0	214.7	212.3
27. Per cent of total steam used by jacket and drain	8.97	17.77	10.17	10.48	10.51	10.14	8.14	9.04	8.41	9.27	9.96	14.31	13.4	14.43	10.87	.....
28. Total revolutions, 1000 rev.	10.09	10.05	10.35	10.42	2.221	2.148	6.512	15.46	15.94	33.18	32.27	43.12	44.87	29.77	41.75	27.02
29. Water pumped per revolution, lb.	5613	5613	6777	6778	6295	5930	2137	6780	6781	6603	6607	1992	1991	5932	2132	6296
30. Total water pumped, million lb.	56.64	56.39	70.16	70.60	13.98	2.74	13.92	104.8	108.1	219.1	213.2	8.589	8.934	176.6	89.00	170.1
31. Total water pumped, million gal.	6.795	6.765	8.416	8.468	1.676	1.527	1.668	12.57	12.96	26.32	25.59	1.029	1.071	21.16	10.67	20.39
Hourly Quantities																
32. Main condensate per hour, 1000 lb.	11.49	10.36	9.76	9.708	7.481	7.147	6.070	10.29	10.63	9.112	8.775	1.646	1.600	7.950	8.138	.....
33. Jacket and drain water per hour, lb.	1133	2238	1105	1137	879	806.5	537.8	1022	975.4	919.0	953	265.7	256.6	1128	972	.....
34. Dry steam used by unit per hour, 1000 lb	12.62	12.60	10.86	10.84	8.36	7.954	6.608	11.31	11.60	9.917	9.571	1.882	1.780	9.040	8.937	8.825
Heat Data																
35. Heat units above 32 deg. fahr. per lb. dry steam throttle, B.t.u.	1275	1278	1267	1262	1269	1265	1260	1264	1263	1262	1256	1194	1194	1195	1195	1195
36. Heat of liquid above 32 deg. fahr. per lb., temp. of jacket and drain, B.t.u.	175.9	129.7	178.3	178.9	158.6	179.0	157.8	154.7	151.3	158.3	153.0	157.1	154.8	192.2	151.7	.....
37. Heat of liquid above 32 deg. fahr. per lb. main condensate leaving final heater, B.t.u.	62.83	167.1	57.31	54.92	58.4	53.32	64.48	71.53	67.97	72.26	76.31	71.87	80.6	.....	.....	.....
38. Total heat, units above 32 deg. supplied in steam hr., million B.t.u.	16.10	16.10	13.76	13.69	10.61	10.06	8.327	14.29	14.65	12.51	12.02	2.257	2.150	10.58	10.75	10.20
39. Total heat units above 32 deg. fahr. returned in jacket and drain per hr, 1000 B.t.u.	199.2	290.2	197.0	203.5	139.4	144.4	84.86	158.0	147.5	145.9	146.2	41.91	39.79	215.9	147.9	.....
40. Total heat units above 32 deg. returned in main condensate per hr. 1000 B.t.u.	7221	1730	5592	533.2	437.2	381.1	391.4	735.8	722.2	650.2	650.2	118.5	129.2	531.7	468.1	.....
41. Total heat units above 32 deg. returned in jacket, drain and main condensate per hr., 1000 B.t.u.	9213	2021	756.2	736.7	576.6	525.4	476.3	893.8	869.7	796.1	796.4	160.4	169.0	733.4	615.4	.....

TABLE 3 (*Continued*) — TEST DATA

Engine	A.C.	A.C.	A.C.	A.C.	Allis	Kilby	Holly	Holly	Holly	Holly	Holly	Holly	Holly	Kilby	Holly	Allis
Column	No. 4 1	No. 4 2	No. 3 3	No. 2 4	No. 1 5	No. 1 6	No. 1 7	No. 6 8	No. 7 9	No. 2 10	No. 3 11	No. 4 12	No. 5 13	No. 1 14	No. 1 15	No. 1 16
42. Net heat units consumed by unit per hour, million B.t.u.	15.18	14.08	13.00	12.95	10.03	9.536	7.850	13.40	13.78	11.72	11.22	2.096	1.981	9.846	10.14	.....
43. Per cent of net units (heat) consumed by jacket and drain water	8.2	18.26	9.25	9.51	9.73	9.19	7.56	8.46	7.87	8.68	7.56	13.44	12.40	11.26	9.99	.....
Speed																
44. Revolutions per minute	21.03	20.93	21.57	21.70	18.51	17.9	27.13	21.47	22.14	23.04	22.41	59.89	62.31	20.67	28.99	18.72
45. Piston speed, ft. per min	231.3	230.2	237.3	238.7	197.4	190.8	217.1	236.2	243.5	230.4	224.1	239.6	249.2	220.5	231.9	199.8
Power																
46. Ft-lb. of work done per hour, million ft-lb.	2670	2659	2240	2220	1734	1510	1332	2188	2266	1773	1678	283.9	300.8	1464	1503	1428
47. Water horsepower	1349	1343	1131	1122	875.5	762.8	672.7	1105	1114	895.7	847.4	147.1	151.9	740.0	759.2	721.5
48. Hydraulic efficiency, per cent	97.98	98.26	.....	.....	.....	.....	.....	.....	.....	98.9	95.7	.....	.....	.....	.....	.....
49. Mechanical efficiency, per cent	97.35	96.47	.....	.....	.....	.....	.....	.....	.....	93.85	95.11	.....	.....	.....	.....	.....
50. Combined hydraulic and mechanical efficiency per cent	95.38	94.79	96.71	97.48	92.82	85.12	91.24	96.23	96.45	92.69	90.91	95.67	95.70	96.28	96.32	.....
51. Average indicated plunger, horsepower	1377	1366	.....	.....	.....	.....	.....	.....	.....	906.8	886.4	.....	.....	.....	.....	.....
52. Average indicated horsepower	1414	1417	1170	1151	943.2	896.1	737.3	1119	1156	966.2	932.0	154.3	158.8	768.0	788.2	.....
53. Horsepower lost in mechanical friction	37.53	50.61	.....	.....	.....	.....	.....	.....	.....	59.45	45.60	.....	.....	.....	.....	.....
54. Horsepower lost by water friction, etc.	27.80	23.19	.....	.....	.....	.....	.....	.....	.....	11.07	39.05	.....	.....	.....	.....	.....
55. Total horsepower losses	65.33	73.80	38.5	28.99	67.72	133.3	64.59	13.4	11.8	70.52	84.65	7.22	6.83	28.6	29.0	.....
56. Per cent of total i.hp. developed in hp. cylinder	41.14	41.54	42.8	43.05	36.66	37.9	43.70	41.80	41.30	39.72	40.45	35.27	36.33	32.46	37.3	.....
57. Per cent of total i.hp. developed in i-p. cylinder	24.08	28.14	25.3	23.47	27.72	23.3	22.65	31.78	32.15	32.63	31.65	32.89	33.64	31.21	34.35	.....
58. Per cent of total i.hp. developed in h-p. cylinder	34.78	30.32	31.9	33.48	33.63	38.8	33.65	26.42	26.55	27.65	27.90	31.84	30.03	36.33	28.25	.....
Capacity																
59. Rate of pumpage, million gal. per 24 hours	20.38	20.29	25.25	25.40	20.11	18.32	10.01	25.14	25.91	26.32	25.59	2.059	2.142	21 16	10.67	20.35
60. Rated capacity, million gal. per 24 hr	20	20	25	25	20	20	10	25	25	25	25	2	2	20	10	20
61. Per cent of rated capacity developed	101.9	101.5	101.0	101.6	100.6	91.5	100.1	100.5	103.6	105.3	102.4	102.9	107.1	105.8	106.7	101.8

Economy and Efficiency																
62. Heat units consumed per w.hp.-hr., 1000 B.t.u. ..	11.25	10.49	11.50	11.55	11.46	12.50	11.67	12.13	12.04	13.08	13.24	14.15	12.87	13.30	13.37	13.55
63. Heat units consumed per i.hp.-hr., 1000 B.t.u....	10.73	9.94	11.12	11.26	10.64	10.64	10.65	11.98	11.92	12.13	12.04	13.53	12.31	12.80	12.88	.....
64. Thermal efficiency of entire unit, per cent.....	22.61	24.27	22.14	22.04	22.21	20.36	21.81	21.00	21.10	19.47	19.23	18.00	19.78	19.14	19.05	18.80
65. Thermal efficiency of steam cylinders, per cent. .	23.71	25.60	22.89	22.61	23.93	23.91	23.90	21.24	21.35	20.98	21.14	18.85	20.68	19.80	19.78	.....
66. Ratio of thermal effy. of unit to that of ideal Rankine cycle	72.78	77.22	70.29	69.27	71.16	64.22	71.17	72.10	73.50	78.20	766	66.1	74.8	69.7	73.5	.....
67. Ratio of thermal efficiency of steam cylinders to that of Rankine cycle.....	76.30	81.46	72.63	71.06	76.61	75.43	77.99	73.0	74.4	83.3	84.2	69.2	78.2	72.5	76.3	.....
68. Pounds of steam consumed per w.hp.-hr., lb.....	9.36	9.38	9.60	9.67	9.55	10.43	9.82	10.23	10.14	11.07	11.29	12.75	11.71	12.23	11.72	11.81
69. Steam consumed per i.hp.-hr., lb...	8.93	8.89	9.29	9.43	8.86	8.875	8.96	10.11	10.03	10.26	10.27	12.20	11.21	11.77	11.34	.....
Duty																
70. Ft-lb. of work per million heat units.....	175.9	188.7	172.2	171.4	172.8	158.4	169.7	163.3	164.5	151.4	149.6	140.0	153.9	145.9	151.2	146.0
71. Ft-lb. of work per 1000 lb. steam, million ft-lb....	211.5	211.0	206.2	204.8	207.4	189.9	201.6	193.5	193.4	178.8	175.4	155.3	169.0	162.0	168.2	167.4
72. Duty per million B.t.u. correct. per test curves, million ft-lb.....	173.8	186.0	171.3	171.1	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
73. Duty per million B.t.u. correct. per Rankine cycle, million ft-lb.....	172.9	184.8	170.4	171.0	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
74. Guaranteed duty per million B.t.u., million ft-lb..	170.0	.....	170.0	170.0	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....

large improvement in the future of any one type must call for a radical departure from the present trend of development.

16 The foregoing comparison of records made by different types of units emphasizes the relatively small horsepower of the pumping engine. The authors consider that the development which has led to such high efficiency in water-works steam pumping engines that require in general a maximum not over 1500 hp., has been largely the result of the practice, instituted by water-works engineers, of making economy tests on all installations.

17 The influence of this requirement of economy tests for practically all installations is also seen in the development of very efficient small-capacity pumping engines of the horizontal crosscompound Corliss type. The economy characteristics of this type are shown in Fig. 3. This requirement has also influenced steam-

turbine development through the application of turbines for driving centrifugal pumps through reduction gears, as is manifested by the use of special highly efficient turbines for such service.

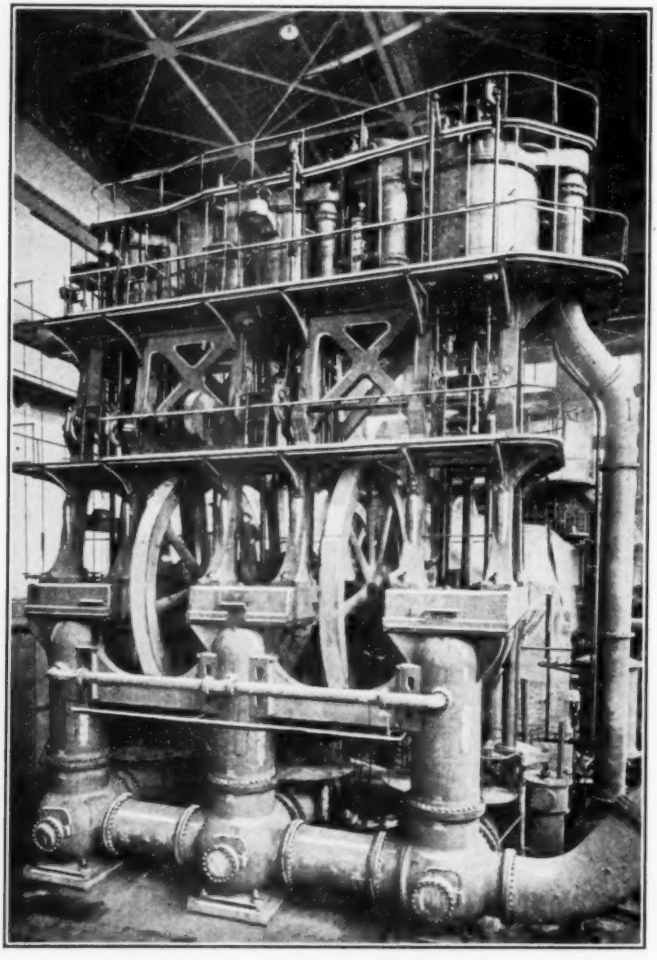


FIG. 4 ALLIS-CHALMERS ENGINE NO. 3 WITH LAGGING REMOVED

18 The rebuilding of Division Station which resulted in the economies referred to included the installation of three new Allis-Chalmers units, two of 25,000,000 gal. per day capacity against 250 ft. head and one of 20,000,000 gal. per day capacity against

380 ft. head. The specifications for these units called for a duty for each of not less than 170,000,000 ft-lb. of work for each million heat units consumed when supplied with steam at 200 lb. gage pressure and 100 deg. fahr. superheat and with only exhaust steam used for feedwater heating.

19 The duty specified indicates the advance that had been made in water-works economy as it was based on tests of Holly engines Nos. 2 and 3 in 1904 and Holly engines Nos. 6 and 7 in 1912 at Kirtland Pumping Station, due weight being given for the greater available heat in the steam under Division Avenue Station conditions as compared with Kirtland steam conditions, and is higher than any of the records which have been cited in Table 2.

20 The thermal efficiency given in Table 2 of 20.95 per cent for the Wildwood unit was obtained by the use of a total of five feedwater heaters, four of them taking steam from the engine between the throttle and the exhaust nozzle of the last cylinder. The Division Avenue units were equipped with an exhaust-steam feedwater heater in accordance with the specifications and another taking steam from the second receiver, but, as has been mentioned, the guaranteed economy for the Division Avenue units was to be based on the use of the exhaust heater only.

21 Tests in addition to those covering the acceptance trials with both heaters in service were made, and one of these is included in Table 2 under date of October 15, 1918.

22 The results of the tests of the new Division Avenue units are given in detail in the first four columns of Table 3. The other columns present the results of the official tests of all of the nine other vertical triple-expansion pumping engines in service in the Cleveland Water Works.

23 The economy of all the units covered in the test results in Table 3 is indicative of the efficiency of this type of engine when built in a wide range of sizes — the smallest being Holly engines Nos. 4 and 5 of 155 hp. and the largest, Allis No. 4 of 1400 hp. — and operated under greatly different steam conditions, and is also indicative of the excellence to which the design has been brought by the different builders.

24 The thermal efficiency of the new Division Avenue units, together with the high average of all of Cleveland's units, and the fact that this type of engine was giving a thermal efficiency of more than 20 per cent by the year 1900, broadens the interest in the results given. For this reason the last item in Table 2 has been added to

express the economy of the unit in the usual central-station term of "B.t.u. per kilowatt-hour" and in Fig. 2 the relation between the water works duty, expressed in foot-pounds of work per million B.t.u. consumed, and the central-station term mentioned, is shown graphically.

25 The official test economy records of all of Cleveland's vertical triple-expansion pumping engines, including the two 155-delivered-hp. units, have been summarized graphically in Fig. 5,

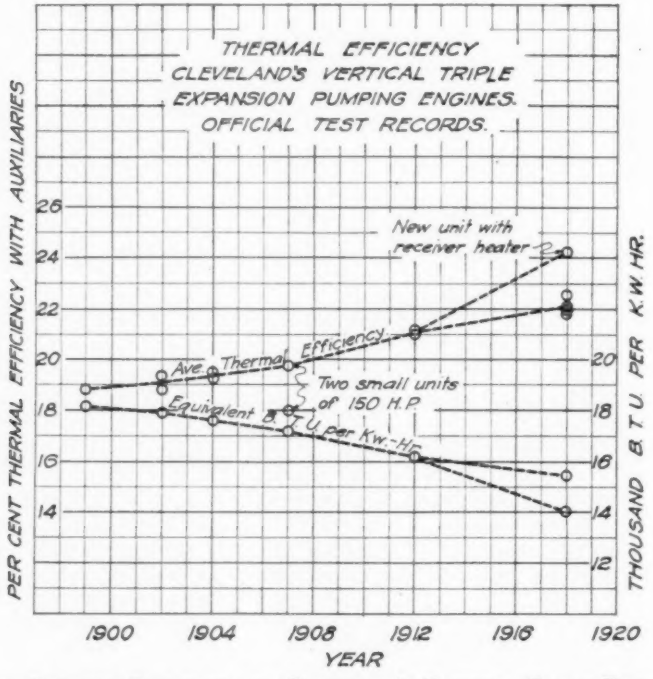


FIG. 5 THERMAL EFFICIENCY OF CLEVELAND'S VERTICAL TRIPLE-EXPANSION PUMPING ENGINES (OFFICIAL TEST RECORDS)

which shows the thermal efficiency and the equivalent B.t.u. per kw-hr. plotted as a function of the year in which the test was made. These records are evidence that the high efficiencies in Table 2 were not peculiar to those engines only, but rather represented good water-works practice.

26 The performance of Allis engines Nos. 2, 3, and 4, previously referred to can be taken as applying in general principle

to the vertical triple-expansion type of pumping engine, and the following interesting developments which were brought out by an analysis of the test results bear on certain characteristics the authors believe are worthy of special consideration.

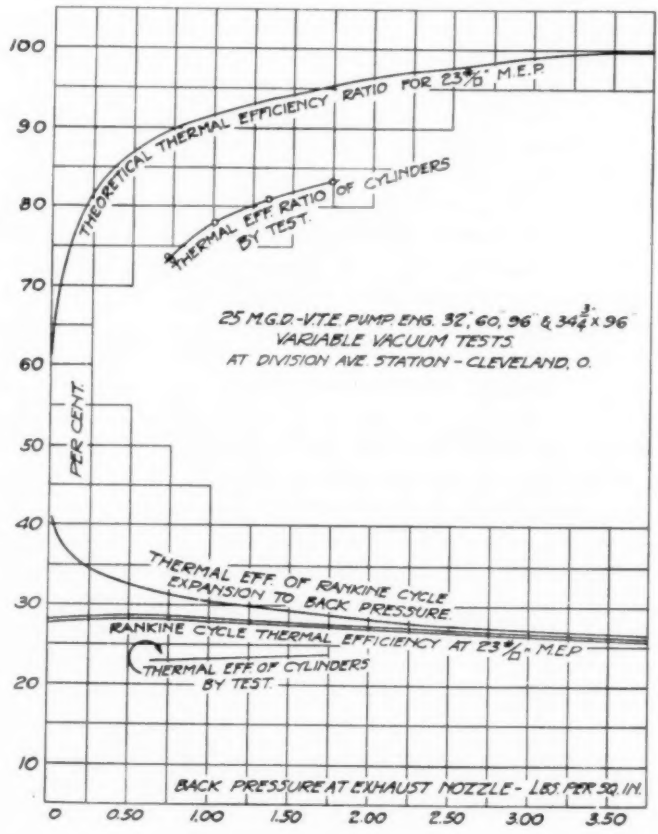


FIG. 6 VARIABLE VACUUM TESTS AT DIVISION AVENUE STATION, CLEVELAND, OHIO.

27 As the major loss in efficiency ratio is due to incomplete expansion in the low-pressure cylinder, it follows that the efficiency ratio should increase rapidly if this loss is reduced by increasing the exhaust pressure. This conclusion is confirmed by the results of a series of variable vacuum tests plotted in Fig. 6. These tests

show that the efficiency ratio of the steam cylinders increased from 73 per cent at 0.725 lb. per sq. in. (28.52 in. vacuum) to 83 per cent at 1.75 lb. per sq. in. (26.44 in. vacuum). The thermal efficiency of the Rankine cycle has also been plotted for comparison with the theoretical thermal efficiency of a constant mean-effective-pressure (or incomplete expansion) cycle having a m.e.p. of 23 lb. per sq. in. This m.e.p. approximates the loading during the tests being discussed. The theoretical thermal efficiency-ratio line on the diagram shows the effect of incomplete expansion on the efficiency ratio of a constant m.e.p. cycle and illustrates how closely the characteristics of the actual engine conform to those of the ideal.

28 The form of the efficiency-ratio exhaust-pressure curve in Fig. 6 is characteristic of the type of engine as well as of the particular unit tested. This characteristic is shown by the tests of seven Cleveland engines plotted in Fig. 7 in which the vacuum varied over a wide range.

29 The efficiency ratio of the high-pressure and intermediate-pressure cylinders combined, which operate, in effect, as a non-condensing compound, is 83 to 85 per cent, with the high-pressure cylinder alone attaining an efficiency ratio of 90 per cent. These high efficiencies result from the loss, due to incomplete expansion, which is so large in the low-pressure cylinder, being reduced to a minimum for the two higher-pressure cylinders, and from the use of an amount of superheat in the high-pressure cylinder which results in the steam being exhausted at approximately dewpoint.

30 The bleeding of approximately 8 per cent of the steam used from the second receiver at about atmospheric pressure, and its utilization for heating feedwater, increases the thermal efficiency of the unit approximately 7 per cent and is accomplished with no increase in steam consumption, for the reason that the loss of available energy of 3 per cent in the bled steam is offset by an increased efficiency of the cylinders due to the resulting better distribution of the work between them. This alteration in the efficiency characteristics is better visualized by considering that when the engine is operated without second-receiver bleeding, the intermediate-pressure cylinder is considerably underloaded and the low-pressure cylinder is heavily overloaded and that these conditions of cylinder loading are bettered by the withdrawal of steam from the second receiver for heating purposes.

31 The performance of the different elements of the steam end which enters into the characteristics summarized, can be estimated

with accuracy from the test data for unit No. 4, and such an examination also discloses the effect of the second-receiver feedwater heater used during the October 15 test. This feature may be of general interest at present because of the attention the subject of stage heating of feedwater is receiving from central-station engineers. The steps followed in analyzing the tests of October 14 and 15, 1918 are described below.

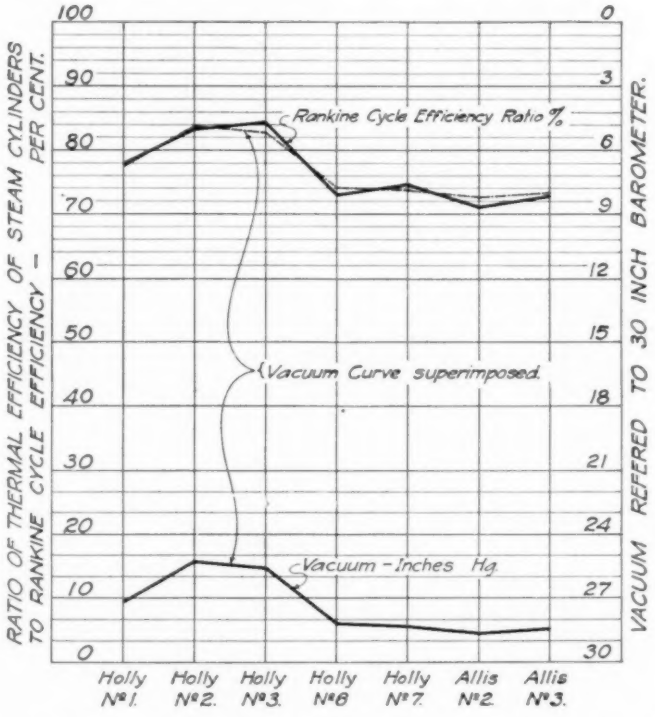


FIG. 7 VARIATION OF RANKINE-CYCLE EFFICIENCY RATIO WITH VACUUM ON SEVEN DIFFERENT ENGINES

32 The efficiency ratio of the high-pressure cylinder and jacket is obtainable from the indicated horsepower, the total steam used, the initial steam pressure and superheat and the first receiver pressure, which is the exhaust pressure for that cylinder. The average quality of the steam leaving the high-pressure cylinder and jacket with correction for estimated radiation is therefore known, but because of the arrangement of steam flow (see Fig. 8) in which the condensed

steam from the high-pressure jacket is discharged direct to the second receiver without being measured, the quality of the steam in the first receiver is not readily estimated. The high-pressure and intermediate-pressure cylinders' economy is estimated from the sum of their indicated horsepowers, the total steam used, the initial steam pressure and superheat and the second-receiver pressure. As all the steam is brought together again in the second receiver, the quality at this point, with allowance for radiation, is determined with accuracy.

33 The receiver volumes are very large and they become effective separators, as is evidenced by the large percentage of drain water discharged from the second receiver. An assumption of the complete separation in the second receiver affords a means of esti-

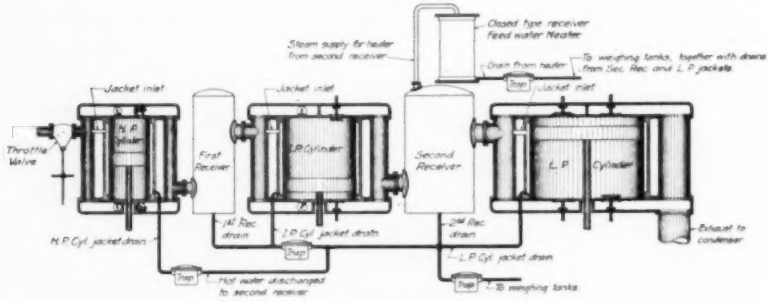


FIG. 8 DIAGRAMMATIC SECTION OF STEAM END, NEW DIVISION AVENUE UNITS

mating the performance of the low-pressure cylinder and jacket, and any error involved in such an assumption has very little effect on computed efficiency ratio.

34 From the test data of October 15 (including the temperature rise of the condensate through the second receiver heater) the amount of steam withdrawn from the receiver for condensate-heating purposes is readily determined.

35 The economy of the different cylinders during the October 14 and 15 tests and the distribution of work between the cylinders are given in Table 4, with sufficient detail to illustrate the methods used in arriving at the different results.

36 The results of the analysis of these two tests have also been plotted on the Mollier diagram in Fig. 9.

37 The remarkable flatness of the efficiency curve of the vertical triple-expansion type of pumping engine under a great variation of

conditions is shown by the curves in Fig. 10, in which the percentage of duty is plotted against variable speed and variable head. The loss in economy is only 5 per cent at half load and 6 per cent at half speed, with flat characteristics on the overload side also.

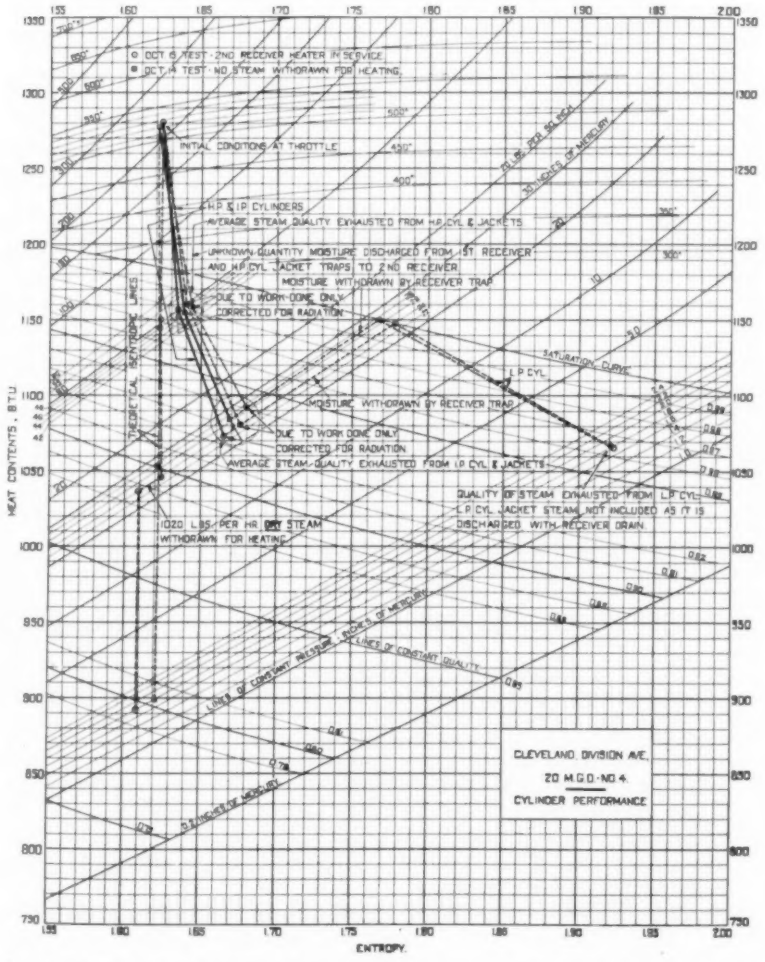


FIG. 9 MOLLIER DIAGRAM OF OCTOBER 14 AND 15 TESTS ON NO. 4 ENGINE

38 In view of the high efficiencies attained by this type of unit under all the operating conditions of steam pressure, superheat, vacuum, speed and load, it is felt that the general features of cylinder

TABLE 4 ECONOMY ANALYSIS OF NO. 4 UNIT, DIVISION AVENUE STATION

(Cylinders 36 in., 68 in. and 108 in.; stroke 66 in.)

Test date (1919)	Oct. 14	Oct. 15
Steam pressure, lb. abs.	221	222
Superheat, deg. fahr.	130	136
1st receiver pressure, lb. abs.	48.2	50.4
2d receiver pressure, lb. abs.	13.25	11.71
Exhaust pressure, lb. abs.	0.885	0.855
ENERGY AVAILABLE, RANKINE CYCLE:		
To entire engine, B.t.u. per lb.	378	381
To high-pressure cylinder, B.t.u. per lb.	132.5	130.5
To high- and intermediate-pressure cylinders, B.t.u. per lb.	224.5	234.5
To low-pressure cylinder, dry steam initially, B.t.u. per lb.	169.5	163.0
To low-pressure cylinder theoretically, B.t.u. per lb.	153.5	146.5
INDICATED HORSEPOWER:		
Entire engine	1417	1420
High-pressure cylinder	582	590
High- and intermediate-pressure cylinders	924	990
Low-pressure cylinder	493	430
PERCENTAGE OF POWER PER CYLINDER:		
High-pressure cylinder, per cent	41.14	41.50
Int.-pressure cylinder, per cent	24.08	28.14
Low-pressure cylinder, per cent	34.78	30.32
STEAM WITHDRAWN FROM SECOND RECEIVER FOR HEATER:		
Temp. rise of condensate through heater, deg. fahr.	.....	106.3
Latent heat at 2d receiver pressure, B.t.u. per lb.	.....	977
Calculated refrigeration of condensed steam in heater, deg. fahr.	.....	100
Heat given up in heater per lb. supplied, B.t.u.	.....	1077
Condensate pumped through heater, lb. per hr.	.....	10,359
Heat absorbed by condensate, B.t.u. per hr.	.....	1,100,000
Weight of steam supplied heater from receiver, lb. per hr.	.....	1020
Weight of steam from 2d receiver, low-pressure cylinder jacket and receiver heater, lb. per hr.	1133	2238
Weight of steam from 2d receiver and low-pressure cylinder jacket, lb. per hr.	1133	1218
Weight of condensate from condenser, lb. per hr.	11,492	10,359
ECONOMY OF HIGH-PRESSURE CYLINDER:		
Weight of total steam supplied high-pressure cylinder, lb. per hr.	12,625	12,597
Steam used per i.hp-hr., lb.	21.70	21.35
Efficiency ratio	0.885	0.915
Average quality at exhaust of high-pressure cylinder and jacket, including radiation	0.9817	0.9827
ECONOMY OF HIGH- AND INTERMEDIATE-PRESSURE CYLINDERS:		
Weight of total steam supplied cylinders, lb. per hr.	12,625	12,597
Steam used per i.hp-hr., lb.	0.830	0.853
Average quality at exhaust of cylinders in 2d receiver, including radiation	0.9325	0.9234
ECONOMY OF LOW-PRESSURE CYLINDERS:		
Weight of total steam entering 2d receiver, lb. per hr.	12,625	12,597
Average quality of steam in receiver, including radiation loss, lb. per hr.	0.9325	0.9234
Moisture discharged from receiver, assuming complete separation, lb. per hr.	852	966
Dry steam withdrawn by receiver heater, lb. per hr.	.....	1020
Net dry steam supplied low-pressure cylinder and jacket, lb. per hr.	11,772	10,612
Dry steam used per i.hp-hr., lb.	23.9	24.65
Efficiency ratio of low-pressure cylinder	0.628	0.634
Calculated weight of steam supplied cylinder jacket, lb. per hr.	280	253

TABLE 4. — *Continued*

EFFECT OF SECOND RECEIVER HEATER:		
Total steam used by high- and intermediate-pressure cylinders, lb. per hr.	12,625	12,597
Energy available to high- and intermediate-pressure cylinders, B.t.u. per lb.	224.5	234.5
Dry steam withdrawn for 2d receiver heater, lb. per hr.	.....	1020
Total steam used by engine from 2d receiver, lb. per hr.	12,625	11,577
Theoretical quality in 2d receiver	0.900	0.887
Theoretical energy available to low-pressure cylinder, B.t.u. per lb.	153 5	144
Theoretical total energy available to engine	47,800,000	46,280,000
Difference in energy available between tests resulting from use of receiver heater and from different steam conditions, per cent	.....	3.18

construction and arrangement, the type of valve gear, and the system of steam flow through the engine will be found of interest.

39 The steam cylinders are jacketed, a jacket for each being supplied with steam at the initial pressure for that cylinder.

40 The high-pressure cylinder has Corliss admission and exhaust valves. The intermediate-pressure cylinder has Corliss admission valves and single-beat poppet exhaust valves. The low-pressure cylinder has single-beat poppet admission and exhaust valves. The valves for all cylinders are located in the cylinder heads with the poppet valves flush with cylinder-head surface when closed. The arrangement and types of valves are indicated on the diagrammatic section, Fig. 8. The types of steam and exhaust valves used, Corliss and single-beat poppet, insure tightness when good workmanship and a detailed design that minimizes distortion under temperature changes are employed. The location of all steam valves in the cylinder heads reduces the cylinder clearance to a minimum, as may be judged from an inspection of Table 5.

41 The slow speed and great dimensions of the steam cylinders and receivers result in the exposure of a large cylinder and receiver surface per 1000 lb. of steam used, but as a result of the thorough covering with a high-grade non-conducting material and the enclosing sheet-metal lagging, the loss of radiation is very small. An estimate of this loss, using generally accepted coefficients of heat transmission, indicates that not more than 1 per cent of the total steam is condensed by radiation.

42 Fig. 8 also outlines the arrangement of jacket-supply steam and the disposition of the receiver and jacket drains. The condensed steam from the high-pressure cylinder jacket is discharged through a trap to the second receiver space. The condensate drain from all

the first receiver spaces and from the intermediate-pressure cylinder jacket is likewise discharged through a trap to the second receiver. All of the condensed steam from the second-receiver low-pressure cylinder jacket, and the second-receiver feedwater heater, when it is in service, is discharged through a single trap to atmospheric pressure. The temperature of the water discharged to the atmosphere does not exceed 200 deg. fahr.

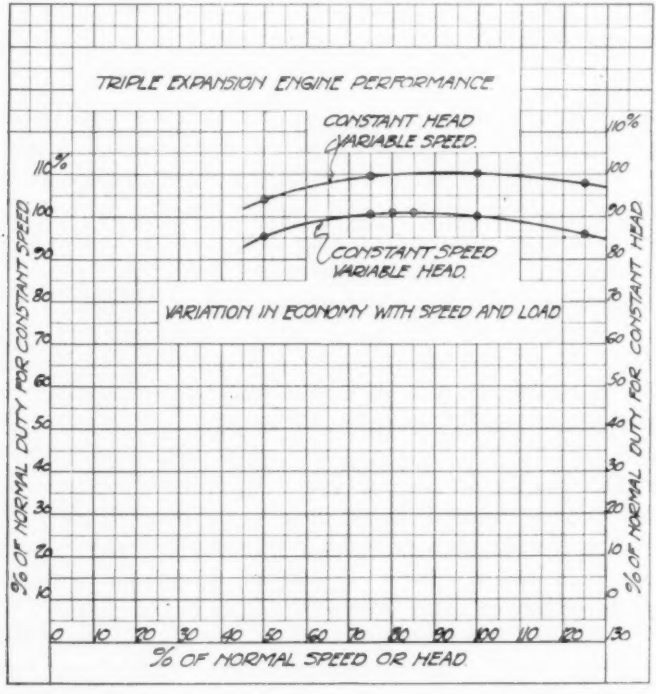


FIG. 10 TRIPLE-EXPANSION-ENGINE PERFORMANCE: VARIATION IN ECONOMY WITH SPEED AND LOAD

43 The steam exhausted from the low-pressure cylinder is condensed in a water-works type surface condenser (the steam flowing through the condenser tubes in two passes) located in the main suction pipe directly adjacent to the pump chambers.

44 The condensate is removed from the condenser by a single-acting bucket-type wet air pump driven direct from the low-pressure plunger head and discharges into a tank from which it is pumped

through the exhaust-steam heater to the station feedwater heater by a single-acting condensate pump direct driven from a main-pump plunger head. The exhaust-steam heater is located in a by-pass connection of the main exhaust pipe just ahead of the condenser.

45 The high efficiencies, thermal and mechanical, of the vertical triple-expansion pumping unit make the matter of test accuracy important. It is possible that for the determination of the amount of work delivered per unit weight of steam this type of machine is inherently the best suited for accurate measurements. The amount of work delivered depends upon two factors, the quantity of water pumped and the head pumped against. The weight of water can be computed from the volume pumped and its temperature, which can be readily measured to within a single degree. The volume of water pumped can be measured by a venturi meter, by volumetric

TABLE 5 AVERAGE CYLINDER CLEARANCES, DIVISION AVENUE UNIT NO. 4

Cylinder	Diameter, inches	Stroke, inches	Cylinder Clearance Volume, per cent of piston displacement
High-pressure	36	66	1.50
Intermediate-pressure ...	68	66	0.50
Low-pressure	108	66	0.25

measurement in a reservoir, and by plunger displacement to within 1 per cent or less of the true amount. The Division Avenue units delivered the water pumped through venturi meters. It has been found on carefully conducted tests that the measurement by pump displacement would check within the accuracy of the meters as evidenced by results in which the meter measurement was slightly greater than the plunger displacement.

46 The head pumped against can be measured by gages and mercury columns to within a fraction of 1 per cent. This measurement during the Division Avenue test was made with a mercury column for the suction pressure and an accurately calibrated bourdon gage for the discharge pressure. Both the mercury column and the gage were directly adjacent to the gage taps into the water mains. The bourdon discharge gage stood free and not subject to any distortion or warping due to gage-board fastenings.

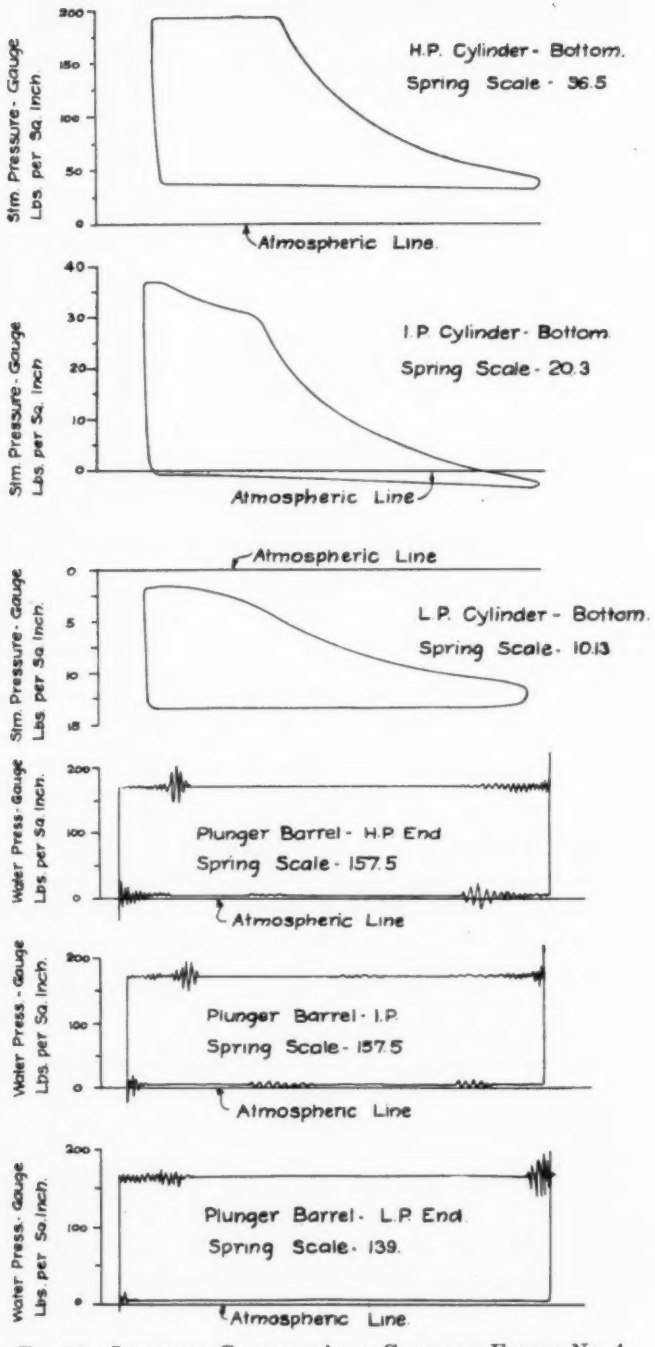


FIG. 11 INDICATOR CARDS OF ALLIS-CHALMERS ENGINE NO. 4
TAKEN OCTOBER 15, 1918

47 The measurement of the work delivered during certain of the Division Avenue tests was checked by a measurement of the indicated horsepower of the pumps and the computation of the hydraulic efficiency. The hydraulic efficiency is listed in Table 3.

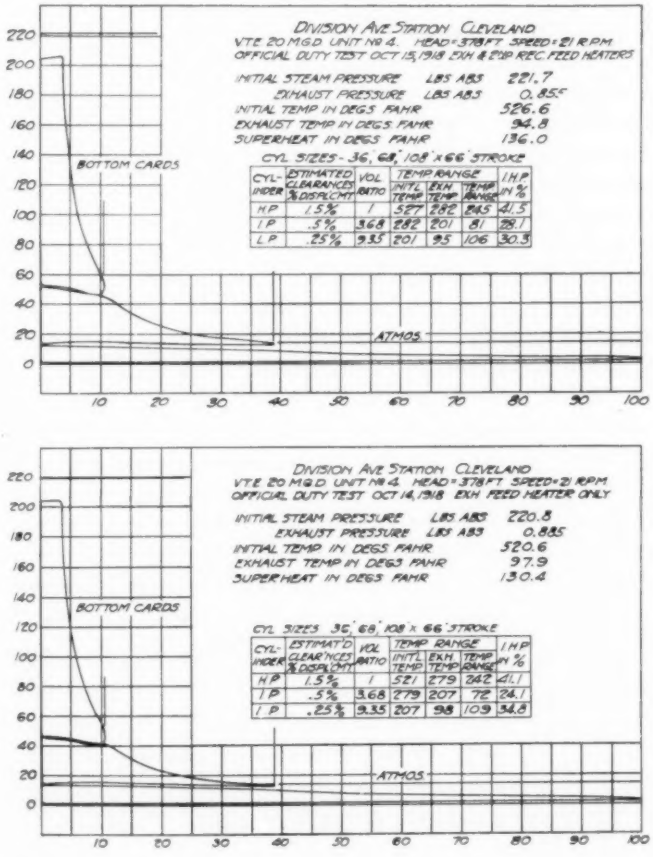


FIG. 12 COMBINED INDICATOR CARDS, DIVISION AVENUE UNIT NO. 4

The measurement of the indicated horsepower of the steam cylinders furnishes a further check on the delivered work, because with the high combined mechanical and hydraulic efficiencies listed in Table 3 as item 50, any considerable error would produce unreasonable results. The great number of measurements which serve as independent checks to the calculated efficiencies practically precludes

the introduction of either deliberate or accidental errors of appreciable magnitude.

48 The measurement of the weight of steam used is accomplished with precision because the relatively small weight used per hour permits the use of accurate portable platform scales that can be readily checked with the standard weights. As in the Division Avenue tests, being portable they can be located close to the discharge of the condensate pumps and the trap handling the jacket and receiver drain, eliminating possibility of pipe-line leakage or open connections. In addition the self-contained feature of the unit makes the isolation of its piping system simple and effective. The water-works type of condenser and its moderate dimensions combined to prevent condenser leakage. The tests in Table 3 represent economies obtained with no deduction from the steam used for condenser or other leakage, as leakage tests indicated that the condensers were tight.

49 The measurement of initial steam pressure, vacuum, superheat, and other quantities is accomplished with the same accuracy with this type of apparatus as for others. The Division Avenue test measurements which apply in all essentials to all other tests tabulated were preceded by accurate dead-weight calibration of bourdon gages and indicator springs and the checking of thermometers and the barometer used. Outside-spring indicators were used on the high-pressure cylinders for superheated steam and the indicator drum was driven from accurate pantagraphs mounted substantially and forming part of the permanent engine equipment.

DISCUSSION

GEORGE A. ORROK. The authors have set before the Society a pumping engine test which in point of economy has exceeded the latest record in our files by a little over three per cent, and the latest reported test of a reciprocating engine by about two per cent.

It is interesting to compare the progress in pumping-engine design by contrasting the first pumping engine which was built and which used about 100 lb. of coal per water horsepower with the test of this engine, which uses practically 1 lb. of coal for each water horsepower.

The only point, to the writer's mind, in which this paper falls short compared with the other papers reporting tests is that the

present authors use the Mollier diagram to show the performance of the engine. While the Mollier diagram may be useful in steam-turbine or engine work, it does not begin to show the things it should. The temperature-entropy diagram is a better and simpler method of showing what takes place in the machine. I think the authors should add a temperature-entropy diagram for the purpose of comparison with the other papers.

GEORGE H. GIBSON. Will not the authors submit some information on the money economy of this type of pumping engine as compared with the geared turbine-driven centrifugal pump?

It is a common statement in paleontology or biology that forms become most complex, perfect and specialized just as they are about to pass out. The present reciprocating triple-expansion engine is the result of a long course of highly refined developments, under the spur of bonuses based on duty guarantees, and a great deal of credit is due to the engineers who have worked out the various details and refinements. However, the perfection of the centrifugal pump, or rather, the perfection of the double-helical, speed-reducing gear, by cutting first-cost in half, has removed the ground from under the tremendous, ponderous, although highly efficient, reciprocating pumping engine. The writer believes he does not err in saying that large pumping stations, such as Cleveland, Philadelphia, Montreal, Chicago, St. Louis and other cities, will probably put in very few reciprocating engines in the future.

C. HAROLD BERRY. It is very interesting to me to notice that the economy of this machine is running as low as 14,000 B.t.u. per kw-hr., and comparing that with the figures presented for the Colfax Station[1] on an entirely comparable basis, this pumping engine seems to be a little more economical than the turbine room of that plant, so that the boiler conditions being equal, the economy of the two should be substantially the same. This being the case, it is hard to see how, from a strictly thermal standpoint, the turbine-driven, geared, centrifugal pump could compete with the triple-expansion engine. The writer presumes that the cost of the reciprocating engine outfit must be greater than the cost of the turbine-driven, geared, centrifugal pump set.

The writer wishes to say that he is greatly impressed with the manner in which this test is presented. It seems to him to be an excellent and valuable piece of work.

[1] See paper No. 1803 *c*.

E. E. MILLER. The writer has noticed with interest that the additional heat obtained by drawing steam for feedwater heating purposes from the receivers gave results about the same as those which were obtained under Mr. Day at St. Louis in 1912. They made about the same tests, taking steam at that time from two receivers for feedwater heating and obtaining from 7.5 to 9 per cent gain in heat units.

Some few years ago the writer plotted a temperature-entropy diagram for an Allis-Chalmers engine in St. Louis, from data obtained during one of the official tests, which shows up the loss due to expansion in the low pressure cylinder, that is, the fact that it cannot be carried out as far as might be desired as mentioned in Par. 27 of this paper.

A. G. CHRISTIE. The most interesting curves to the writer are those of Fig. 7 in which the vacuum is plotted with the Rankine-cycle efficiency. It will be noticed that the Rankine-cycle efficiency decreases as the vacuum increases, and referring to the data of Table 3 this is generally true. However, the newer engine which the authors tested shows in item 67 of Table 3 an efficiency of 81 per cent, which is very nearly as good as the best efficiency plotted on this curve.

In item 66 of Table 3, for the Allis-Chalmers No. 4 an overall-Rankine-cycle efficiency of 77 per cent is quoted. The writer does not believe any turbine-driven centrifugal pump can approach this figure by a considerable percentage. The first cost of the turbine-driven pump would be considerably less, however. Regarding the comparison of the turbine-driven centifugal with the reciprocating engine pumps, a short time ago the writer visited one of the large pumping engine plants in this country and saw an engine which had not been overhauled for any extensive repairs since it had first been put in service in the 70's. It is very doubtful if there are any turbines built now which will operate for that period of time without being rebuilt and overhauled, and a considerable number of parts renewed.

D. S. JACOBUS. It would appear from the figures given in the paper that the B.t.u. consumption per kw-hr. for the pumping engine was about the same as that secured with large steam turbines. Mr. Miller states that a great deal of loss comes at the low-pressure end of the cycle where the full effect of the expansion is not realized Mr. Orrok, who is very familiar with the temperature-entropy diagram, might tell us whether there would be any advantage from a

theoretical standpoint in combining a reciprocating engine for the high-pressure end of the cycle with a steam turbine operated with the exhaust from the engine.

GEORGE A. ORROK. Taking in a general way the question put by Dr. Jacobus any combination is usually not as good as either part by itself. We have, however, combined the engine and turbine, for instance, the combination at the 59th Street station, which has given remarkably good results. I cannot quote from memory, but it is my recollection that that engine-turbine combination was as good as the best turbine of two or three years' later and was better than the turbine at that particular time. The cost of the outfit, however, if it was to be put in new, would be very great.

As applied to pumping engines, we have not a thermodynamic problem. It is clearly a mechanical problem, dealing with the means of running the turbine at a speed low enough to suit the pump, or else building a pump that can be run at turbine speed.

In the turbine-driven centrifugals used for waterworks, the efficiencies obtained have been something in the neighborhood of 60 to 130 million ft. lb.

Mr. Gibson proposes to put in a reducing gear between the turbine and the pump, but some of us have had enough troubles with gears and do not quite fancy it. We have not come to the point where we can put our trust in the reduction gear end. I understand from naval engineers that the reduction gear is working well; they say they have overcome its incident troubles and that it will run, at least, as long as the prime mover that drives the reduction gear.

I should say, taken by and large, that we can get the best efficiency in pumping water out of a straight reciprocating engine with the proper arrangement for using the heat in the receivers and heaters, and that we cannot do any better than that, with the combination turbine-centrifugal through a gear or without a gear.

I am talking about efficiency. Money economy is a different thing. In the old days, when we bought engines and pumps for fifteen cents a pound a waterworks pumping engine could be bought for somewhere between five and seven cents a pound. Some of the very best did not cost over five cents. Allowing for the rise in the price of pig-iron and steel, and a little something for the rise in labor, I presume you can buy a good pumping engine today for perhaps eight or 8.5 cents a pound. Turbines cost all the way from 50 cents to $2.00 a pound. Centrifugal pumps of the ordinary type

are about the cost of engines. When they are of the extraordinary type, beyond 50 per cent efficiency, you must pay for the efficiency. On, say, a 60,000- or 70,000- or 80,000-gallon per minute pump, which would be somewhat comparable with this, a pump of 70 per cent efficiency probably could be had for 12 or 14 cents a pound, but in order to get a really good efficiency, the pump should have an efficiency of about 90 per cent, and it would probably be necessary to pay double the price per pound for such a pump. Some day we shall be building 95 per cent centrifugal pumps. Perhaps in those days we can afford to use them for waterworks engines.

Engines can be built that will run almost any weight per horsepower and, as a general rule, the higher the weight per horsepower the engine is, the cheaper the engine per horsepower.

FRANK L. FAIRBANKS. Having had a great deal of experience with centrifugal pumps in pumping calcium-chloride brine, and salt water, and with comparatively large pumps, and knowing that in this service the initial efficiency of the pump extends over a period of less than six months, and that the average useful life of the impeller in salt water is less than a year, I would like to ask what is the depreciation in efficiency of centrifugal pumps of the larger types in waterworks service.

In the case at Buffalo some years ago, tests showed that the efficiency of the centrifugal pump had dropped radically within six months, and within a year and a half the drop was practically prohibitive. That city then decided on putting in a new pumping equipment, and went back to the reciprocating pumping engine. It would be interesting to know just what the impellers of the centrifugal pumps do in this service. I know that where there is gritty water, or wide variation in head, the centrifugal pump is not well adapted for the work, in as much as in either case it has been demonstrated that initial efficiencies are not maintained.

GEORGE H. GIBSON. In the City of Minneapolis Water Works, there are two 24-in. centrifugal pumps driven by an electrical motor. The City pays on a duty basis, and for that reason frequent tests are run to determine how much water flows through the venturi meters and the head to which it is delivered. The efficiency of the pumps was, when first put in use, 85.97 per cent. This efficiency has not fallen off in the course of about two years.

The principal cause of falling off in efficiency of centrifugal

pumps, outside of the erosion of the impeller, is usually leakage through the wearing rings. The rings in this case are removable, they run at close clearance, the leakage path is tortuous and there is very little leakage and comparatively little wear of the rings, the original rings being still in place. Water that would seriously corrode the pump would be unfit for consumption.

The turbine-driven, geared, centrifugal pump occupies a space of approximately one-fifth or one-sixth the cubical space occupied by a triple expansion engine. From the plunger to the top of a reciprocating engine may be 40 feet or more. This makes not only a tremendous cost in labor and material, but also a difference in cost of foundations and building. These large reciprocating engines require heavy building walls and heavy cranes to handle them.

As to the cost of upkeep, someone has questioned whether gears would run a number of years. The first large gear-driven pump for waterworks service was put into use at Pittsburgh about ten years ago. That unit is still running satisfactorily; there has never been any change in the gears, nor have there been any repairs of any kind made to the turbine. A reciprocating pump, operated for that length of time, might be kept in service, but in that time it would have had its valves and packings renewed over and over again. Leakage in the reciprocating pumping engine, as shown by papers which have been published, may be anywhere from 4 to 40 per cent. The writer has been told by a man connected with the New York Waterworks, that when they found a pump was slipping more than four per cent, they overhauled it, and a force of men was kept on the job continually making measurements of delivery and checking them against the displacement.

None of the waterworks gears with which the writer is familiar have given any trouble, and there are something like 50 large units now in operation. There are a number of them in the City of Cleveland — and the writer believes that Mr. Quayle could give some interesting first-hand information regarding the economy of operation of these machines. There are other units in Philadelphia, Chicago, St. Paul, Indianapolis, Montreal, Toronto and other large cities throughout the country.

Leonard A. Day. As supplementary data to the excellent and thorough paper by Messrs. Quayle and Brown, we will submit the results of a test on a 20-million-gallon duty, vertical triple-expansion Holly pumping engine at our Bissell's Point Station.

This is a more recent test than those referred to in Table 2 of the paper, and is worthy of record.

High service pump No. 13 has cylinders 34, 64 and 98 inches in diameter with 66 in. stroke, and is comparable to the Cleveland Holly pumps Nos. 6 and 7. The test was of 24-hour duration, and was the acceptance test of the unit. Test results are as follows:

TABLE 6 TEST DATA, HIGH-SERVICE PUMP NO. 13, ST. LOUIS WATER DIVISION

Steam pressure, lb. per sq. in.	154.43
Barometer, in	29.795
Superheat (as read by thermometer), deg. fahr.	101.58
Vacuum, in.	28.0
Temperature of exhaust, deg. fahr.	96.88
Reading of pressure mercury column, ft.	301.55
Elevation pump discharges, ft.	92.0
Average temperature of mercury column, deg. fahr.	76.13
Elevation water in wet well, ft.	95.31
Diameter high-pressure plunger, in.	32.62
Diameter intermediate-pressure plunger, in.	32.616
Diameter low-pressure plunger, in.	32.615
Stroke of high-pressure plunger while running, in.	65.875
Stroke of intermediate-pressure plunger while running, in.	65.828
Stroke of low-pressure plunger while running, in.	65.906
Water pumped in 24 hours (actual plunger displacement), gal.	20,576,500
Water pumped in 24 hours (Venturi meter readings), gal.	20,610,000
Weight of condensate, lb.	230,145
Weight of first receiver drain, lb.	1,883
Weight of second receiver drain, lb.	8,299
Weight of jacket drain, lb.	11,763
Total weight of steam, lb.	252,090
Weight of steam per hour, lb.	10,503
Temperature of water pumped, deg. fahr.	58.45
Temperature of condensate, deg. fahr.	61.79
Temperature of first receiver drains, deg. fahr.	263.104
Temperature of second receiver drains, deg. fahr.	197.705
Temperature of jacket drain, deg. fahr.	164.297
Plunger leakage per hour, lb. (Specification allowed 100 gal.)	333.00
Absolute steam pressure, lb. per sq. in.	174.03
Superheat (corrected emergent steam thermometer), deg. fahr	105.23
Average value of superheat in duty: 10 deg. equals 1,000,000 ft-lb.	
Pressure by mercury column (corrected for temperature) ft.	301.013
Total head pumped against (95.31 − 92 = 3.31; 301.013 − 3.31 = 297.703) ft.	297.703
Gallons per revolution	714.437
Weight water pumped per revolution, lb.	5,957.7
Average revolutions per hour	1,200.00
Duty, ft-lb. per 1,000 lb. steam without correction for excess in superheat, $\frac{1{,}200 \times 5{,}957.7 \times 297.703}{10.5037}$	202,628,616.00
Deduct for 5.23 deg. of superheat, net duty, lb. per 1,000 lb.	202,105,616.00
Heat per lb. superheated steam, B.t.u.	1,256.00
Total heat delivered to engine, B.t.u.	316,625,040.00
Heat per pound condensate, B.t.u.	29.87
Total heat in condensate, B.t.u.	6,874,431.00
Heat in one pound of first receiver drains, B.t.u.	231.8
Total heat in first receiver drains, B.t.u.	436,479.00

Heat per pound second receiver drains, B.t.u.	165.63
Total heat in second receiver drains, B.t.u.	1,374,563.00
Heat per pound jacket drains, B.t.u.	132.16
Total heat jackets, B.t.u.	1,554,598.00
Total heat rejected, B.t.u.	10,240,071.00
Heat consumed by engine, B.t.u.	306,384,969.00
Total ft-lb. work done in 24 hr.	51,080,404,697.30
Duty ft-lb. of work per million B.t.u. consumed by engine; $\frac{24 \times 1,200 \times 5,957.7 \times 297.703}{306,384,969}$	166,719,796.00
Water horsepower developed	1,074.91
Steam per water-horsepower-hour	9.77

The duty in foot-pound per B.t.u. on test was 166.7; by the installation of an exhaust heater this duty was raised to 169.3. If it is assumed that the ratio of the thermal efficiency of the unit to that of the ideal Rankine cycle does not vary for an increase in the steam conditions from those in the above test, to those of A.C. pump No. 4 of the Cleveland tests, then the increase in duty would be in direct proportion to the increase in the Rankine cycle efficiency, an increase of 4.8 per cent. The resultant duty will be $169.3 \times 1.048 = 177.5$, a duty comparable with that of the Cleveland A.C. pump No. 4. The overall thermal efficiency of the unit, including the exhaust heater, is 21.8 per cent for the steam condition of the above test, and 22.8 for the steam conditions of A.C. pump No. 4 of the Cleveland tests.

Not only does this type of unit develop a high efficiency but it maintains its high efficiency over a period of years. A 20-million-gallon daily, vertical, triple-expansion Allis high-service pump at Bissell's Point Station developed the same duty as that of its acceptance test 16 years previous.

I. H. Reynolds. The present paper recalls the circumstances leading up to the design of this type of engine.

The earliest steam engines were used for pumping mines in England and in the development of the reciprocating engine since that time, the pumping engine has always been the leader in point of economy. For mine pumping, the cylinders were naturally placed vertically and on account of the desirability of getting the steam cylinder away from the mine shaft a "beam" was employed, placing the steam cylinder at the opposite end from the pump and it is a curious instance of the persistence of type, long after the original reason for its use had disappeared, that until thirty-five years ago practically all pumping engines were built with some form of beam

or bell crank and it was not until the vertical triple-expansion engine became firmly established that the use of beams and levers was definitely abandoned.

The earliest pumping engines were "direct acting" and the power engines which soon followed were practically the pumping engines with a crank and flywheel added and all early steam engines, even those on the first locomotives, had vertical steam cylinders and "walking beams."

The economy of the early pumping engine was measured by the foot-pounds of water raised per hundred weight (112 pounds) of coal burned and from this custom was established the so-called "duty" test, and until a comparatively recent date the duty was based on coal rather than on steam consumption, the engine builder usually supplying the boilers and being responsible for the entire plant.

Variation in the quality of coal, efficiency of boilers and skill of firemen gradually lead to assuming an evaporation of ten pounds of water per pound coal and finally the coal standard was abandoned altogether so that duties are now expressed in terms of steam or B.t.u., the latter method equalizing the heat conditions due to variation in steam temperatures and permitting the engine efficiency to be measured independently of the boiler plant.

The early pumping engines, having either single plungers or being driven from single cranks, produced heavy pulsations which were partially taken care of by air chambers and as the engines usually discharged into reservoirs, moderate pulsations were not objectionable, but with the coming of direct-pressure pumping, the necessity arose for an engine capable of giving a flow of water free from appreciable pulsations.

The Worthington duplex pump, in which one plunger was supposed to start on its stroke before the opposite plunger had stopped, partially eliminated the pulsations, and the Holly-Gaskill compound engine, which was brought out in the late '70s or early '80s, was built with two double-acting plungers, thus giving four deliveries per revolution, and was especially adapted for direct pumping and was the first crank and flywheel engine built to a standard design, but like practically all previous crank and flywheel engines, the use of a beam was a feature of the design.

The problem of finding a simple form of pump which should give a steady delivery of water was first solved by the E. P. Allis Company of Milwaukee in 1881 or 1882 when a proposal was submitted to the City of Nashville for a vertical three-cylinder com-

pound pumping engine, but for certain reasons the old type of beam engine was chosen by that city.

In the engine proposed, there were three steam cylinders, one high pressure and two low pressure, which stood at the main floor level, and the crank shaft and flywheel were located above the steam cylinders while three single-acting pumps were located below and the pump plungers directly connected to the piston rods of the steam cylinders.

The main shaft had three cranks set at angles of 120 degrees. (the forerunner of the modern triplex pump) which gave an overlapping delivery of water, resulting in practically a straight discharge line, while the pumps were of the simplest possible construction.

In 1883 the same company built for the city of Alleghany, Pa. (now a part of Pittsburgh — "North Side"), two, three-cylinder compound pumping engines of the type proposed a year or two earlier for Nashville and these engines were, as far as the writer knows, the first three-cylinder pumping engines ever built.

The Alleghany engines were very satisfactory from all standpoints and gave a duty of 107,000,000 ft-lb. per 1,000 pounds of steam, the pressure being 100 lb. (saturated) and the test being made by Professor Greene of Rensselaer Polytechnic Institute.

In May 1886 the city of Milwaukee advertised for a 6,000,000-gal. compound engine and the writer having in mind the success of the Alleghany engines and being further influenced by his experience as a marine engineer, designed the present type of vertical engine which differs from the Alleghany machine principally in having the steam cylinders above and the crankshaft below, the steam end following the usual marine type of construction in general outline.

At this time triple-expansion engines had been installed on a few ocean steamers and were attracting considerable attention on account of increased economy over the compound engines previously in use. The marine engines were using 160 lb. steam pressure, and although the steam pressure specified at Milwaukee (80 lb.) was considered much too low for best economy, it was decided at the suggestion of Edwin Reynolds, superintendent of the Allis company, to offer a triple-expansion engine as an alternate proposition against a three-cylinder compound, and the writer made the design, using cylinder sizes calculated by B. V. Nordberg, who at the time was one of the Allis company's engineers.

The duty guaranteed for the triple was 115,000,000 ft-lb. per

100 lb. anthracite coal as against a guarantee of 100,000,000 for a three-cylinder compound, and on account of this difference in economy the proposal for the triple was accepted by the city engineer, George H. Benzenberg, whose faith in the new type of engine was amply justified, as on the official test it actually gave a duty of 118,000,000 ft-lb. per 100 lb. coal or about 130,000,000 per 1,000 lb. of steam, which figures were unprecedented for an engine operating with such low steam pressure.

About six months after the contract was taken for the Milwaukee engine, a triple-expansion engine was designed in England under the supervision of W. B. Bryan, chief engineer of the East London Water Works Company, and was installed in one of its plants.

The engines were built simultaneously, although the Milwaukee design had a priority of about six months, and neither party had any knowledge of the work being done by the other.

The English engine was built from marine patterns, having a piston valve on the high-pressure and slide valves on the intermediate- and low-pressure cylinders, and while entirely successful from a mechanical standpoint and operating with high-pressure steam, it never approached the economy of the Milwaukee engine with its lower steam pressure, which is accounted for largely by the inferior valve gear and greater cylinder clearances of the English engine.

The East London Water Works Company later installed similar engines fitted with Corliss valve gear, and various English builders adopted the design; and although some of the later machines gave very high economy, they never have quite equalled the best American records.

The success of the Milwaukee engine was so pronounced that the city of Chicago, in 1888, ordered five 18,000,000-gal. engines of the same general design, which machines were guaranteed to give 125,000,000 ft-lb. duty per 1,000 lb. of steam, the pressure being 125 lb.; and the actual duty attained (test by R. W. Hunt & Company) was about 149,000,000.

The installation of the Chicago engines definitely marked the abandonment of all types of beam pumping engines and other triples were rapidly installed in the representative cities of the country. The Snow Steam Pump Company (now Worthington) brought out a horizontal triple-expansion engine for oil pumping, which was practically the Milwaukee engine laid on its side and a little later the same company adopted similar construction for a

horizontal compound engine, which is now the recognized standard "opposed" type, compound pump.

A triple-expansion engine installed at the Milwaukee Water Works in 1890 gave a duty of over 154,000,000-ft-lb. per 1,000 lb. (saturated) steam at 125 lb. pressure being the equivalent of 11.68 lb. steam per i.hp. or 217.6 B.t.u. per i.hp. per minute.

The test was conducted by Prof. R. C. Carpenter of Cornell University and was the subject of a paper by the late Doctor Thurston[1] which provoked a great deal of discussion as the economy shown was so greatly in advance of any previous records with steam at this pressure.

Similar engines in St. Louis Water Works with 135 lb. steam (saturated) gave as high as 177,000,000 ft-lb. duty with 21 per cent thermodynamic efficiency, and an engine in Boston with 185 lb. (saturated) gave a duty of 179,000,000 ft-lb. with a thermal efficiency of 21.63.

The present vertical triple-expansion engine retains all of the essential features of the first design and the improvements and economy have been due to refinement of details and especially to the employment of higher steam pressures and temperatures.

For a period of thirty-five years this type of engine has held first place as the ideal pumping engine having the highest economy and reliability and lowest upkeep and depreciation.

Alvin Schaller. The authors make a comparison of the thermodynamic efficiency of four different types of units which is both misleading and erroneous.

For both the pumping engine and the turbo-generator the efficiency is based upon the heat in the steam supplied to these units, but in the case of the Humphrey gas pump and the natural-gas engine the efficiency is based upon the heating value of the fuel consumed.

If we were to assume that the boilers at the Cleveland Pumping Plant were operated with the best obtainable modern practice, the thermodynamic efficiency of the plant would lie between 19 and 20 per cent, a very good performance indeed. It is this figure, however, and not 24.3 which should be used in making a comparison with an internal combustion engine or a Humphrey gas pump.

The Authors. The purpose of the paper as indicated by the title was to set forth the economy characteristics of the triple-ex-

[1] Trans., vol. 15, 1894, p. 313.

pansion pumping engine from data which covered a wide range of conditions and which had been checked sufficiently to determine that the test results were accurate. Every effort was made to present sufficient records to establish the efficiency and characteristics of this one type of machine without introducing the corrections and modifying considerations which are unavoidable when comparisons of different types of machines used for the same service, are made.

The discussion has brought out the intense interest of all in the matter of the relative merits of the engine-driven reciprocating and the turbine-driven centrifugal types of pumps, and Mr. Gibson has mentioned the centrifugal installations in the Cleveland water works. In addition to the twelve vertical, triple-expansion pumping engines, referred to in the text, the city of Cleveland has in service thirteen steam, turbine-reduction-gear-driven, centrifugal pumping units having a total capacity of 625,000,000 gallons per day. The centrifugal installations include the types of pumps used for different heads between 50 ft. and 470 ft., and were all made between the years of 1913 and 1921.

The purchase and operation of the Cleveland centrifugal installations have been supervised by one of the authors and it is the intention to incorporate their performance in a paper which will cover the acceptance tests, and tests made after several years of operation, and which will furnish a basis for a comparison of the centrifugal and reciprocating types of water-works pumps.

The test of the Bissell's Point Holly engine which Mr. Day has presented in Table 6 is a confirmation of both the high efficiencies shown by the vertical triple type in Cleveland and the close agreement of the plunger displacement and venturi meter measurements of water pumped which the authors have mentioned. The difference between the two measurements during the 24-hr. St. Louis tests was about $\frac{1}{6}$ of 1 per cent, an amount that is, in itself, a small proportion of the usual and accepted error of the venturi meter.

The use of the Mollier diagram for indicating graphically the performance of the steam cylinders has certain advantages which to the authors seemed to give it preference over the older (but not usually employed) temperature-entropy chart. All the work and comparisons which are made at the present time involving cycle efficiencies, energy available during isentropic expansion and other such thermal factors are usually at some stage based on the Mollier total-heat-entropy chart. This chart is so extensively used by all engineers that its application to the analysis of test data from re-

ciprocating engines makes it most desirable to visualize the approximate expansion line on the Mollier plane. The single advantage that the temperature-entropy diagram has of showing by an area, the work done after release, does not warrant its continuance in the presentation of test results. The Mollier diagram has also the very practical advantage in the use of standard published charts on which the data can be plotted.

The comments of I. H. Reynolds giving in detail the circumstances which led to the development of the type of pumping engine described in the paper are of especial interest from a historical point of view at this time when the machine is built and used so generally without distinction, by name, as to its origin.

We shall have to take issue with Mr. Schaller's statement that the comparison which we made of thermodynamic efficiencies of four different types of units is both misleading and erroneous. In our opinion the statement is perfectly correct as we state that we are comparing four different types of prime movers and not four different types of power plants, and our statement of the relative thermal efficiencies is therefore entirely correct. The fact that steam engines use heat in the form of steam and the gas engines use heat in the form of gas does not in any way affect the thermal efficiencies of the prime movers as such.

Further, if we assumed the gas engines were operating on producer gas and the producers had the same efficiency as Division Station boilers, which is an entirely reasonable assumption, then not only the thermal efficiencies of the prime movers would be as stated but also the plant efficiencies would be comparative as given.

No. 1812

BOILER-PLANT EFFICIENCY

By Victor J. Azbe, St. Louis, Mo.
Member of the Society

The usual wastes in boiler plants are brought out strikingly by the author by means of tables and curves of boiler performance compiled from a large number of observations.

The object of this paper is to show to what extent these wastes are preventable or can be made to balance each other, and to recommend a standard for boiler operation toward which designers and operators of boilers may aim.

The author considers that there still remains an immense field for experimental research to obtain the necessary data for boiler construction. The requirements of the ideal boiler installation of today are summarized.

THE degree of waste in boiler plants is not generally realized and there is a tendency to judge conditions by a few of the better plants where improvements have been made and whose results are considered as representative of the average-run boiler plants. This is unfortunate and the author ventures the statement that while considerable improvement resulted from the efforts of the U. S. Fuel Administration, the field of opportunity was barely scratched and if further effort is not made the work of the Administration will be greatly minimized.

2 Tables 1 and 2 give particulars of two boiler tests made under ordinary operating conditions, one on a hand- and one on a stoker-fired boiler. During these tests care was taken that actual operating methods were pursued. The results are very bad, but representative of most of the smaller and some of the larger installations. Most tests do not show up such inefficiency, but that is because they are not made under actual operating conditions.

3 Table 3 gives the results of three boiler tests, all made on the same boiler some time apart. The boiler was of the B. & W. type, but poorly set, with a limited combustion chamber of only 0.87

Presented at the Annual Meeting, December 1921, of The American Society of Mechanical Engineers.

cu. ft. per rated boiler hp. The stoker in use was of the old Jones type.

4 During Test A, the aim was to maintain the same operating conditions as existed before any improvements were made; the result was an efficiency of 56 per cent and boiler rating of 102 per cent. After this first test improvements were begun, without, however, in any way altering the stoker or increasing the combustion space.

TABLE 1 TEST ON HAND-FIRED BOILER REPRESENTING VERY UNECONOMICAL CONDITIONS

Date of test	Dec. 2, 1920
Type of boiler	B & W
Grate	Shaking
Kind of coal	Illinois Screenings
Steam pressure gage, lb.	135.6
Temperature of feedwater, deg. fahr.	205.0
Temperature of flue gas, deg. fahr.	568.4
Draft at back damper, in.	0.48
Draft over the fire, in.	0.24
Carbon dioxide, per cent	5.68
Quality of steam, per cent	98.5
Rated boiler hp.	150.0
Rated capacity developed, per cent	105.9
Evaporation, equivalent coal as fired, lb.	4.91
Evaporation, equivalent coal dry, lb.	5.51
Calorific value of coal, dry basis, B.t.u.	11510
Coal analysis:	
Moisture, per cent	10.9
Ash, per cent	15.8
Volatile matter, per cent	35.2
Fixed carbon, per cent	38.1
Carbon in ash, per cent	24.4
HEAT BALANCE	Per cent
Heat absorbed by the boiler	45.31
Loss due evaporation of moisture in coal	1.31
Loss from vapor of hydrogen combustion	5.41
Loss in dry flue gas	28.21
Loss due to carbon monoxide	4.24
Loss due to combustible in ash	5.40
Loss due to heating of moisture in air	0.30
Loss due to radiation, hydrogen, hydrocarbon, etc.	9.22

5 After about six months, results were obtained as represented by Test B; the efficiency was increased to 68.5 per cent and capacity to 140 per cent of rating. The results were not as yet considered satisfactory and efforts for further improvement were made. In all about 30 boiler tests were run under different conditions and with different fuels and one of the last is represented by Test C in Table 3.

TABLE 2 TEST ON STOKER-FIRED BOILER REPRESENTING VERY UNECONOMICAL CONDITIONS

Date of test	Feb. 23, 1917
Type of boiler	Heine
Type of grate	Detroit
Kind of coal	Pocahontas Nut & Slack
Steam pressure, gage, lb.	138
Temperature of feedwater, deg. fahr.	152
Temperature of flue gas, deg. fahr.	761
Draft at back damper, in.	0.98
Draft over fire, in.	0.38
Carbon dioxide, per cent	8.1
Coal burned, lb.	14700
Water evaporated, lb.	96480
Evaporation (actual), lb.	6.56
Evaporation (equivalent), lb.	7.25
Boiler hp. developed per hr., lb.	386.6
Boiler rating, hp.	328
Rating developed, per cent	118
Heat value of fuel, B.t.u.	13950
Boiler efficiency, per cent	50.4

TABLE 3 BOILER TESTS REPRESENTING IMPROVEMENT OF BOILER PERFORMANCE

Type of boiler	B & W		
Type of stoker	Jones		
Kind of coal	Illinois Screenings		
Test [1]	A	B	C
Steam pressure, gage, lb.	130	127	129
Temperature of feedwater, deg. fahr.	180.6	187.0	176.8
Temperature of flue gas, deg. fahr.	612	533	532
Draft at back damper, in.	0.57	0.34	0.11
Draft on fire, in.	0.45	0.24	0.01
Forced draft pressure, in.	3.00	3.52	4.04
Carbon dioxide, per cent	7.6	13.3	14.0
Quality of steam, per cent	0.991	0.991	0.999
Rated boiler hp.	269	269	269
Rated capacity developed, per cent	102	140	185
Evaporation, equivalent, coal as fired, lb.	6.14	8.19	8.43
Evaporation, equivalent, coal dry, lb.	6.95	9.05	9.55
Calorific value of coal, dry basis, B.t.u.	12080	12850	11864
Coal analysis:			
Moisture, per cent	11.7	9.9	11.7
Ash, per cent	13.4	8.9	12.5
Volatile matter, per cent	31.0	32.2	32.5
Fixed carbon, per cent	43.9	49.5	43.3
Carbon in ash, per cent	16.7	19.4	17.2
Efficiency of boiler furnace and grate, per cent	56.0	68.5	78.1
Improvement in economy over Test "A," per cent	.....	22.3	39.5
Increases in capacity over Test "A," per cent	.....	31.4	81.4

[1] Test A indicates conditions when no effort toward economy was made while Tests B and C show gradual improvement under an economy campaign. In all three cases maximum possible capacity was striven for.

This test is very remarkable in several respects, but is correct. In fact, the results are checked by other tests giving equal or even slightly better results.

6 The remarkable features of the results of this test are as follows:

a High boiler efficiency with a relatively inefficient installation and relatively poor coal

b Complete combustion with high-volatile coal and a combustion space of only 0.47 cu. ft. per developed boiler hp., or 0.11 cu. ft. per lb. of coal including the space occupied by the coal

c A draft drop through a 16-section 9-tube (each 16 ft. long) B. & W. boiler of only 0.1 in. at 85 per cent overload

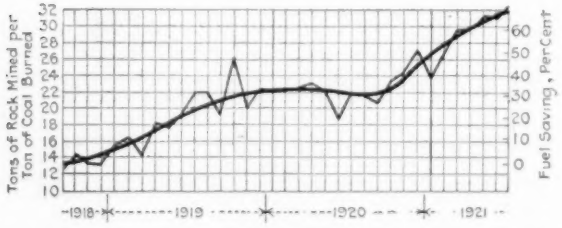


FIG. 1 RESULTS OF IMPROVING BOILER EFFICIENCY IN LARGE MINING PLANT

d A flue-gas temperature of 532 deg. fahr. at 85 per cent overload, in spite of the limited combustion space and flame extension to almost the end of the first pass

e An improvement of 39.5 per cent in economy over Test A

f An improvement of 81.4 per cent in capacity over Test A in spite of the fact that during the first test maximum boiler rating was aimed at.

7 The results from the improvements were manifold. While the plant at first was limited in boiler capacity and operation was often handicapped by low steam pressure, after conditions were improved only half of the boilers usually maintained on the line were necessary, in spite of the increased plant output. It also was possible to reduce boiler-house labor, saving over 16,000 tons of coal per year. Boiler maintenance was also reduced, and if improvements had been begun in time a large part of the money invested in the plant equipment would have been saved.

8 Fig. 1 gives a curve showing improvement in the above plant over an extended period, but taking the power plant as a whole. The present rate of saving is over $100,000 per year, with the additional advantages of a possibly greater plant output, less interruption in power service, and lengthened life of power equipment. All these are made possible by comparatively insignificant costs of improvements, but mainly by careful attention to sound engineering principles and persistence in the effort to obtain the goal set as a standard.

9 Other examples could be enumerated of great improvements being made: A fire-brick plant where fuel consumption was reduced 42 per cent with plenty of opportunity for improvement left; a manufacturing plant operating four boilers, cut out two in the economy campaign and continued to operate with only two boilers, effecting a reduction of fuel consumption of 35 per cent; many plants where it was thought necessary to put in more boilers to carry the load but where it was demonstrated that there were already too many boilers and that improving economy increased capacity.

STANDARD OF ECONOMY

10 The writer considers 90 per cent efficiency as the standard of economy attainable under favorable conditions, this 90 per cent representing boiler and economizer without taking auxiliary power into consideration, however. The requirements for this efficiency are rather severe and with 12,000 B.t.u. bituminous coal, 15 per cent excess air would have to be used, flue-gas temperature would have to be no more than 250 deg. fahr., combustion complete and no carbon in the ash. These requirements are listed in Table 4. The ideal or 90 per cent boiler and furnace efficiency was attained only once, by Henry Kreisinger and John Blizard at the Lakeside Station of the Milwaukee Electric Railway and Light Company.

FUELS

11 With a suitable installation, almost as high an efficiency will be obtained with low- as with high-grade fuels. The only real obstacle to this is the tendency toward a greater ashpit loss with lower-grade fuels. As far as any other factors are concerned, if we analyze them we note that with proper installation their effect is very small. One reason for poor boiler-room efficiency and lack of entire success with low-grade fuels is the tendency of one man to

copy equipment without sufficiently considering the necessary adjustments for adaptation to the different conditions.

12 As a rule low-grade fuel should be burned close to the mines and higher-grade fuel transported. The mines should be especially equipped to burn the very lowest-grade fuels, such as the fine material scooped from under the coal seam by the coal-cutting machine. This dust is often mixed with clay, but can be burned with entire success with proper equipment. That low-grade fuels can be burned successfully is demonstrated by the boiler-test data given in Table 5.

TABLE 4 IDEAL BOILER AND FURNACE PERFORMANCE

Kind of fuel	Bituminous coal
Heat value of fuel, B.t.u. per lb.	12000
Moisture of fuel, per cent	10
Hydrogen in fuel, per cent	3.5
Theoretical air per lb., lb. of coal	8.98
Excess air, per cent	15
Actual air per lb., lb. of coal	10.33
Flue-gas temperature, deg. fahr.	220
HEAT BALANCE	Per cent
Loss due to dry chimney gas	3.15
Loss due to moisture from hydrogen	2.85
Loss due to moisture in coal and air	1.00
Loss due to carbon in ash	1.00
Loss due to incomplete combustion	0.00
Loss due to radiation	2.00
Total loss	10.00
Boiler efficiency	90.00

13 Freight to distant points should not be compared on a tonnage but rather on a heat-value basis. The cost of fuel also should be based on heat value rather than on weight. Buying on heat-value basis, while apparently contrary to the coal operator's interest, actually is not, and once conditions become adjusted and this system becomes general, every one will be better off.

14 As to fuel oil, it should be made an offense against the law to burn crude oil under boiler furnaces; only the use of residue from oil refineries should be permissible and that should be conserved as much as possible for ocean-going steamers and locomotives. Fuel oil is the ideal fuel for boiler purposes. With fair handling it gives high boiler efficiency, high boiler capacity, great load flexibility, low cost of handling and firing, with hardly any disadvantages,

but it should not be burned in localities with a coal mine only a few miles off.

COMBUSTION

15 There are several reasons why properly proportioned furnaces with ample combustion space are necessary, the most important being the prevention of escape of unburned gas and formation of soot, the possibility of operating with the least amount of excess

TABLE 5 BOILER TESTS SHOWING HIGH ECONOMY AND CAPACITY WITH POOR FUEL

Date of test	May 4, 1920	May 5, 1920
Duration of test, hr.	6	6
Object of test	Capacity	Economy
Type of boiler	B. & W.	B. & W.
Type of stoker	Coxe	Coxe
Kind of coal	Yard cleanings	Yard cleanings and Kinkaid storage
Steam pressure, gage, lb.	241.9	245.7
Steam temperature, deg. fahr.	645.7	639.6
Temperature of feedwater entering boilers, deg. fahr.	258.7	264.4
Temperature of feedwater entering economizer, deg. fahr.	138.9	147.8
Temperature of flue gas leaving economizer, deg. fahr.	405.4	381.7
Moisture in fuel, per cent	16.10	17.65
Ash in fuel, per cent	17.14	19.32
Heat value of fuel, dry, B.t.u.	11662	11422
Total amount of fuel consumed, lb.	97935	82929
Water pumped to boiler, lb.	620653	522450
Equivalent evaporation per lb. of dry fuel, lb.	8.634	8.675
Horsepower builder's rating	1220	1220
Per cent of rating developed, including economizer	310	259
Combined efficiency of furnace and boiler, per cent	71.844	73.71
Combined efficiency including economizer, per cent	79.43	81.504

air without incomplete combustion and the prevention of flame entrance among the boiler flues. This last is necessary to obtain low flue-gas temperatures and will be taken up again under that heading.

16 Furnace volumes vary from 1 cu. ft. per lb. of coal burned per hr. down to less than 0.1 cu. ft. in some cases. The lower amounts, of course, do not represent the entire combustion space because combustion under such conditions extends among the tubes, at times for a considerable distance, but this is not what is to be desired. The larger amounts are used with powdered fuel and are necessary because the solid particles burning are relatively large and require a larger combustion chamber for their gasification and combustion than is the case when gasification, and to a certain extent

mixing with air, takes place on a grate. With Illinois coal and a combustion chamber of 0.5 cu. ft. per lb. of coal burned, very good results should be obtained provided the arrangement is correct. Fig. 2 shows test results made by the U. S. Navy Department. The large combustion space is primarily the reason why an efficiency of 66.5 per cent is possible with a rating of 400 per cent. With an ordinary installation and such high rating, flame would be noticeable in the uptake, flue-gas temperature would exceed 1000 deg. fahr. and boiler efficiency be decidedly less than 50 per cent.

17 The aim is to have the gas completely burned and no flame entering the first tube pass. To accomplish this, more than mere furnace volume is necessary. Owing to the tendency of the gas to take the shortest path, the combustion space should be of relatively small cross-section and great length. The cross-section should be

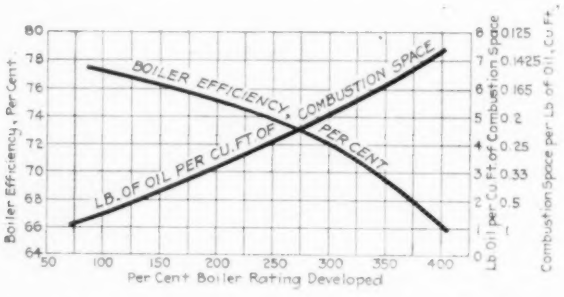


FIG. 2 RESULTS OF IMPROVING BOILER EFFICIENCY AS SHOWN BY TESTS OF U. S. NAVY DEPARTMENT

as uniform as possible, and where it changes it should do so gradually, except where mixing effect is intended. The most effective combustion space is right above the fuel bed where the gas has not had the chance to arrange itself in high- and low-velocity streams, and therefore ample space at this point is most to be desired.

18 The combustion space required varies with the composition of fuel burned and the conditions under which it is burned. It is very apparent that with hand firing larger combustion space will be required than with stokers, from which the volatile matter is given off more uniformly. Excess air is an extremely important factor; temperature also has an important influence, and it is fortunate that as excess air reduces temperature increases.

19 The U. S. Bureau of Mines has contributed a large amount

of very valuable knowledge on this subject but further information taken under more varied conditions is necessary and the Fuels Division of the Society would do well to appoint a committee for the study of this subject.

EXCESS AIR

20 The largest preventable loss in boiler plants is that caused by excess air, and determination of this loss by the measure of the CO_2 and flue-gas temperature has achieved considerable popularity. However, where such popularity has been attained the trouble has been that quite generally there is satisfaction if the fuel is burned with 100 per cent excess air, and even in well-operated plants no better results are obtained on the average. This is because of the tendency of many an engineer to assume statements as correct without sufficient consideration of the circumstances. With the large combustion chambers coming into vogue nowadays, the old assumption that 10 or 12 per cent CO_2 is all that pays becomes obsolete. If we design our furnaces so that flame terminates before gas enters the tubes, or soon thereafter, and we have the boiler heating surface so exposed that the maximum possible amount of heat is radiated to it, then the standard of good performance is the burning of the fuel with 15 or at the very most 25 per cent of excess air, and CO_2 maintained should be around 16 per cent.

21 The point at which excess air enters is very important and should be given greater prominence when boiler-test data are recorded. At present during a test we tend to be primarily concerned with what occurs on the ends, that is, in the furnace and uptake, and few, if any, observations are taken and records made of what occurs between these points. Such items as point of flame termination, relative amount of oxygen, carbon dioxide percentages (which are a true measure of excess air), and temperature drop at various points through the settings, should be carefully recorded and interpreted; then it will be possible to really compare the performance of one boiler with another.

22 A somewhat overlooked but also very important factor affecting boiler-plant economy is "constancy." The percentage of CO_2 may fluctuate during the day from high to low, but if for the whole day a 10 per cent average is taken, it will be found that, for a number of reasons, the efficiency will not be nearly so good as when 10 per cent of CO_2 is constantly maintained. Therefore the average CO_2, when it fluctuates, is an improper measure of the loss but at the same time a good indicator of improper conditions.

HEAT TRANSFER AND FLUE-GAS TEMPERATURE

23 The temperature of the escaping products of combustion determines fuel loss to a great extent, but what is low and what is

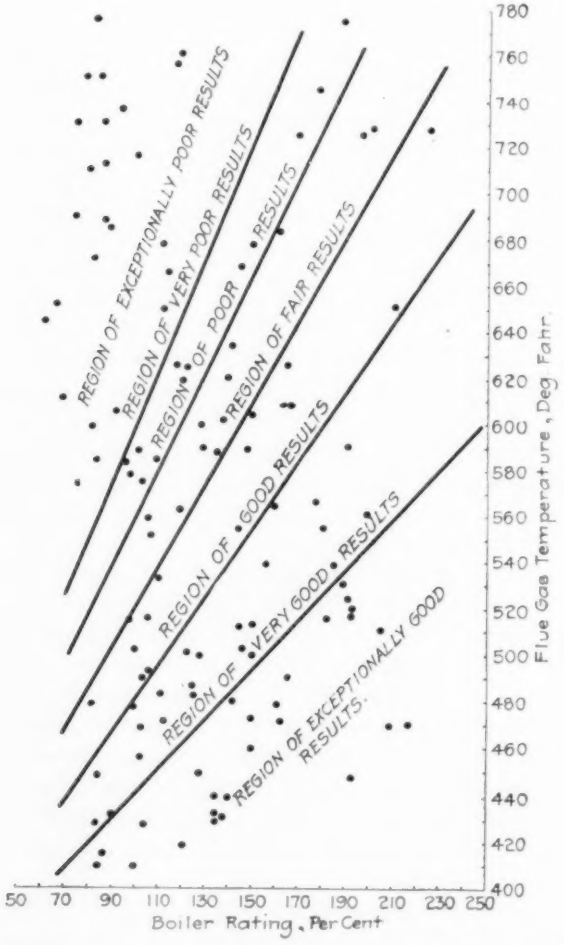


FIG. 3 FLUE-GAS TEMPERATURE REPRESENTATIVE OF GOOD AND BAD RESULTS

high flue-gas temperature? The lack of a suitable measure and standard has in the past prevented proper comparison of results and has also caused a certain lack of incentive for improvement.

The writer, in his effort to evolve a measure, has drawn Fig. 3 based on 150 different boiler tests. Each dot represents the average flue-

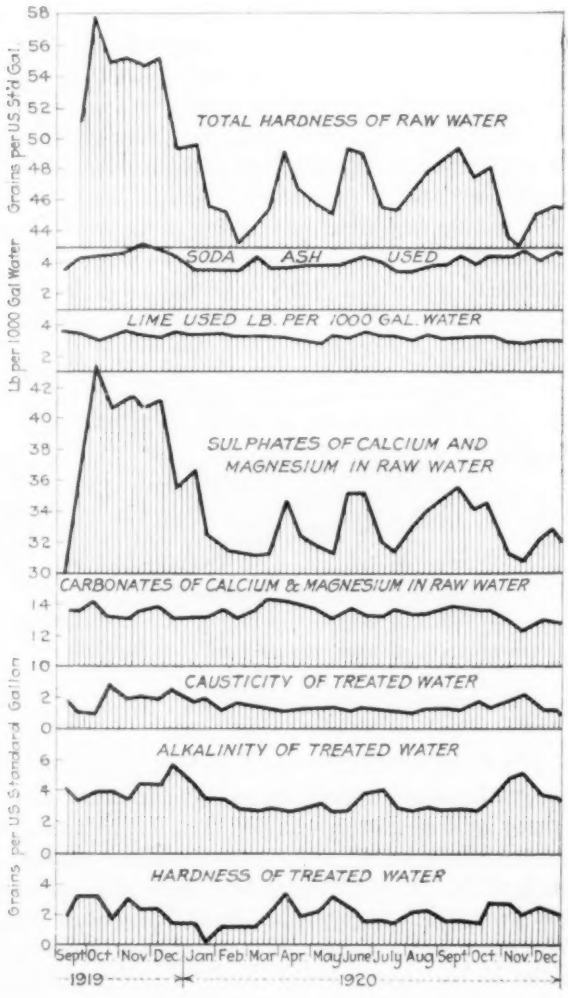


FIG. 4 RESULTS OF A BOILER FEEDWATER TREATING PLANT, HOT SYSTEM

gas temperature at the average rating developed in each test. The space is divided into sections varying from exceptionally good to exceptionally bad results, with the slope of the line following as

closely as possible the increase of flue-gas temperature with the increase of boiler output of a number of boiler-test series.

24 To obtain a low flue-gas temperature, it is necessary that the heating surface be clean. This is self-evident. That the heating surface can be kept clean internally is shown by Fig. 4, which gives graphical results of a boiler feedwater treating plant over a period of 16 months with extremely bad makeup water. It will be noted that the hardness was reduced from an average of 48 to slightly over 2 grains per U. S. gallon. Regarding soot formation on the tubes, a great deal depends upon size of combustion chamber, which if ample will greatly facilitate the keeping of clean surfaces.

25 To absorb the heat, it is necessary to bring the gas in contact with the heating surface. This some boilers do imperfectly. If the surface is clean everything depends upon gas velocity and dis-

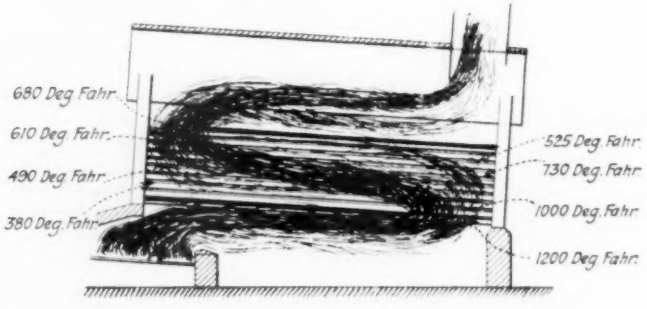


FIG. 5 UNEQUAL GAS-FLOW DISTRIBUTION THROUGH A BOILER

tribution. Fig. 5 shows what happens under certain conditions. In the dead spaces temperatures as low as 380 deg. fahr. were recorded when the flue-gas temperature was 680 deg. fahr. The gas has the tendency to take the shortest cut, a great deal like a stream of water; in fact, much might be learned from the study of water stream flow which could be applied very profitably to boiler-pass design.

26 The cross-sectional area of a boiler pass should be such as to permit the highest gas velocity consistent with the draft available, and also such that the velocity will be constant and not reduce as the gas cools off.

27 Fig. 6 was drawn in an effort to show how boiler heat transfer varies according to different influencing factors. This chart is interesting from many angles and while not exactly correct

for all conditions, it is sufficiently so to permit its use. It is based on the theory that from the actual heat-transfer standpoint we are not concerned with what the furnace temperature is, but only with the maximum theoretical temperature that would have existed had there been no absorption of heat in the furnace. If the boiler absorbed no heat from the furnace, the theoretical temperature corresponding to CO_2 would be obtained very nearly. According to this theory, if CO_2, flue-gas temperature, and rating at which boiler is operated are known, heat transfer can be determined with the aid of the chart. The heat transfer is both by radiation and convection, and the chart is based upon steam temperature of 365 deg.

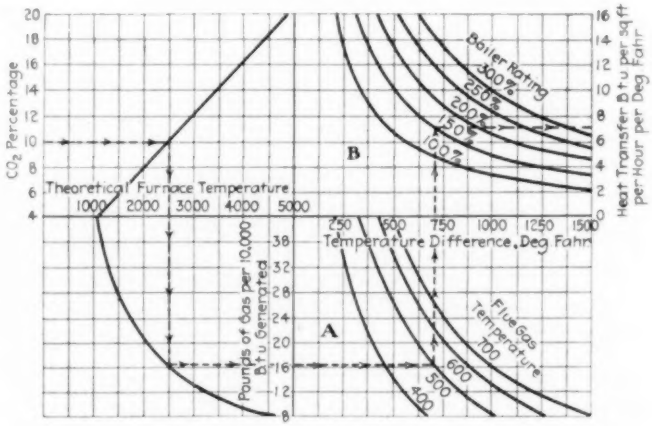


FIG. 6 BOILER HEAT-TRANSFER VARIATION UNDER DIFFERENT CONDITIONS

Based on 365 deg. fahr. steam temperature.

fahr. Since heat-transfer ability is the only possible measure of the efficiency of the boiler itself, it should be apparent that the method should have value in helping to segregate boiler and stoker performance and placing boiler performance on a definite basis of measurement.

28 Fig. 7 demonstrates that with 140 lb. boiler pressure, from 14 to 25 per cent of the total heat used to generate steam is required just to heat the feedwater to 350 deg. fahr. It also shows how the necessary boiler and economizer surface varies with different temperatures of feedwater and that the total boiler and economizer

surface is less than the necessary surface of the boiler without economizer. For maximum economy, counter-current heat-flow effect must be employed in boiler plants.

29 Economizers pay under almost any conditions if they are installed correctly, especially if we take into consideration that they replace more than their own amount of boiler surface. It should be general practice to install smaller boilers which may be forced

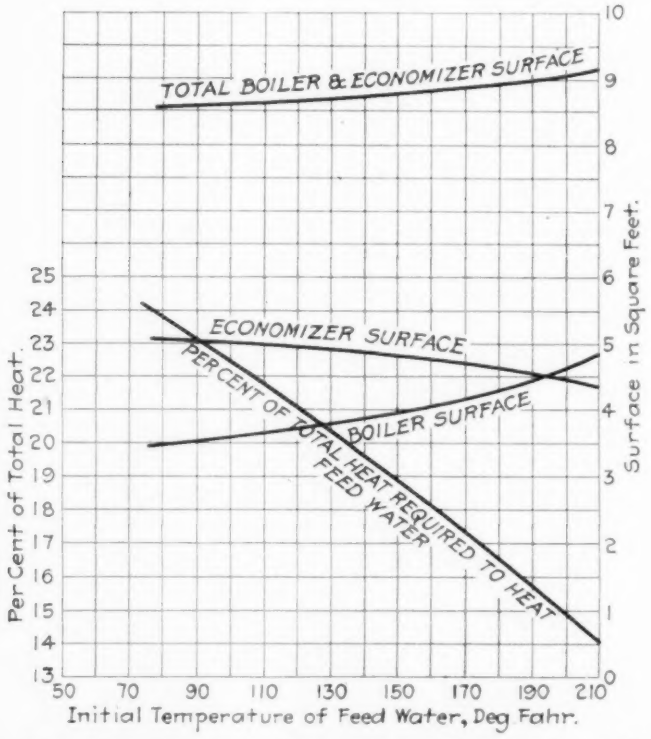


FIG. 7 HEAT REQUIRED TO RAISE FEEDWATER TO TEMPERATURE OF 350 DEG. FAHR.

Based on economizer heat transfer of 3 B.t.u. per. sq. ft. per hr. per deg. fahr.

to greater overloads and the consequently high flue-gas temperatures reduced in economizers. Water should always enter boilers at above 325 deg. fahr. In fact, it is the approach of feed water temperature to steam temperature in the boiler that is a measure of fuel loss resulting from not employing at all or not to a full enough

extent the principle of counter-current heat flow. Fig. 8 applies when there is sufficient heat in the flue gas to heat the water to the temperature of the steam, giving fuel loss for whatever temperature water enters the boiler. With this chart it is possible to tell not only what good a certain economizer installation accomplished but also what good it does not accomplish.

30 Boiler performance should also be judged by the final temperature approach, and it should be kept in mind that it would be manifestly unfair to expect the same efficiency under otherwise equal conditions from a boiler operating at 250 lb. pressure as from one operating at only 100 lb. when the temperature in the first is 67 deg. fahr. higher. The terminal temperature approach of water in boiler and gas in setting at a given rating should always be taken into consideration.

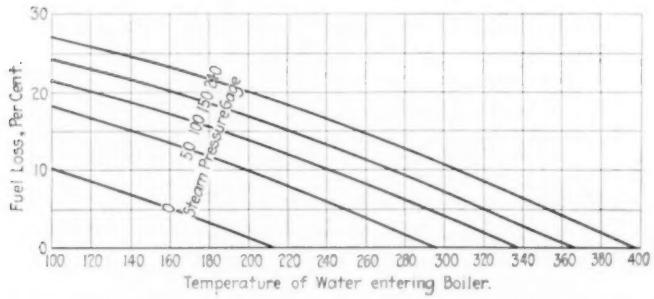


FIG. 8 FUEL LOSS WHEN WATER ENTERS BOILER BELOW STEAM TEMPERATURE

This figure applies only when water could be preheated by waste heat.

31 If a boiler would reduce the temperature of flue gas to the temperature of water in the boiler, the boiler heating surface very evidently would be 100 per cent efficient, no matter what the combined boiler furnace and stoker efficiency would be. Fig. 9 has been drawn in an effort to show the relation of furnace temperature, CO_2, flue-gas temperature, and boiler efficiency for a boiler temperature of 365 deg. fahr. This chart has some limitations, but it nevertheless shows that a boiler as a heat absorber can be 99 per cent efficient.

BOILER CAPACITY

37 When boiler capacity is increased by the burning of more fuel, the following either do or may occur:

a Temperature in combustion chamber increases
b Temperature of gas throughout boiler setting increases
c Temperature difference between water in boiler and gas surrounding tubes and drums increases
d Gas velocity increases
e Heat absorption by radiation increases owing to higher furnace temperature
f Heat absorption by convection increases owing to higher gas velocity and higher temperature difference
g Flue-gas temperature increases
h Dead spaces become more active
i Incomplete combustion increases
j Fuel loss resulting from air in leakage and radiation expressed as a percentage of total decreases
k Excess air decreases
l Fireman's attention usually increases
m Return received from investment increases.

33 There are several factors in this list that counteract each other, some increasing and others reducing efficiency, but as a balance, within limitations, increased boiler capacity is more economical, especially when the installation is correctly made. Curves published showing the variation of boiler efficiency with capacity and reduction of efficiency beyond 150 per cent of rating should not be given too much attention, since everything depends upon conditions which cannot be taken into consideration. Table 4 shows an increase in efficiency from 56 to 78.1 per cent, primarily because capacity was increased from 102 per cent to 185 per cent of rating. In many plants important economies were effected by the simple means of taking off some of the boilers and making the fireman give sufficient attention to the remaining ones to maintain steam pressure.

34 It is the writer's experience that the very large majority of plants possess and operate too many boilers, at times twice as many as necessary. The cry is "Give us plenty of boilers," and the result is waste in investment and operation.

35 But it is not really the fault of the boiler that efficiency is low at low ratings. The fault lies either with the setting, the stoker, or the fireman, and is usually due to them all. Fig. 10 shows graphically results obtained during a series of boiler tests. The remarkable feature of these tests was that boiler efficiency continued to increase with reduced boiler rating, and at 56 per cent of rating

was 84 per cent while at 120 per cent it was only 80.5 per cent. But while such performance is unusual, it should not be so. However, to get such results requires careful, constant and expert attention, which is unusual.

36 Since high boiler efficiency at low ratings is more difficult to obtain than at high ratings, it follows that owing to this and also

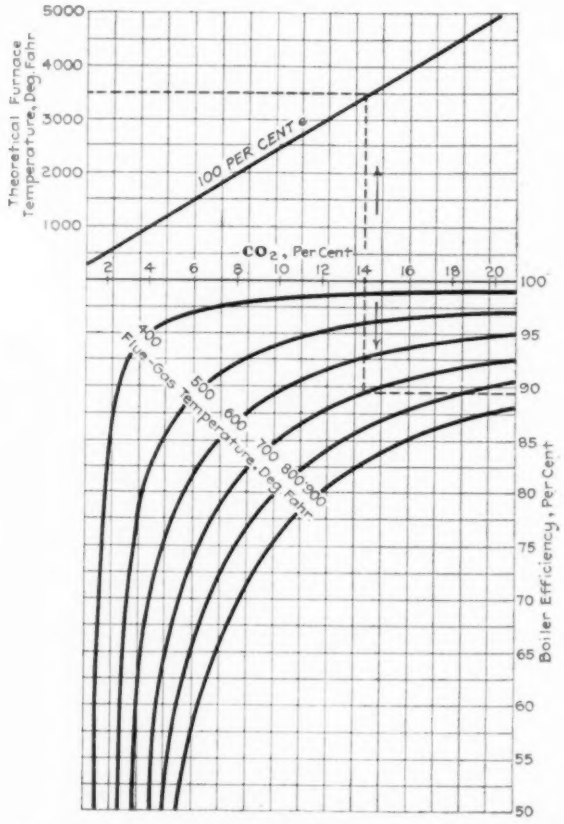


FIG. 9 EFFICIENCY OF A BOILER AS A HEAT ABSORBER

for investment reasons, boilers should be installed designed for high overloads and high and constant efficiency over a considerable capacity range. Especially in plants where load varies a great deal the boilers should be able to take the peaks; even if these peak loads are somewhat uneconomical they are to be preferred to boilers loafing

the greater part of the time. As a rule good boilers, well installed, should be operated at between 150 and 200 per cent of rating, and when a peak is of short duration 300 per cent of rating is permissible. Ratings below 150 per cent should not be permitted, except under unavoidable circumstances. The proper rating at which to operate a boiler depends upon circumstances. In Fig. 3 one boiler has a flue-gas temperature of over 700 deg. fahr. at 100 per cent rating, while another has only 510 deg. fahr. at 200 per cent. Of course the first is entirely unfit for any overload capacity, while the second is good for as much as 300 per cent of rating.

INVESTMENT

37 A new up-to-date installation designed for high capacity and efficiency does not cost initially much more, if any more, per

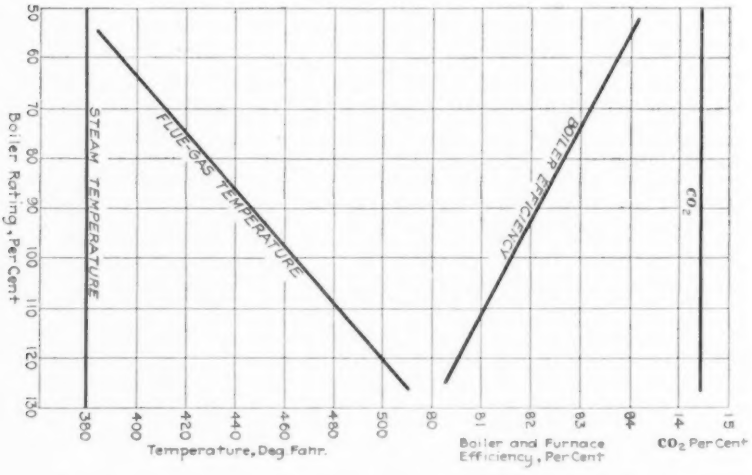


FIG. 10 STIRLING BOILER PERFORMANCE, 712.9 HP. OIL-FIRED

Note increase of efficiency with low ratings and approach of flue-gas temperature to steam temperature.

horsepower developed than the ordinary run of wasteful plants. The subsequent operation of such a plant will be cheaper and so the fuel saving is almost entirely a net saving. It is certainly better to pay twice the amount per rated horsepower for an installation having 100 per cent overload capacity at high efficiency, and economical in

labor, than only half this amount for a wasteful installation capable only of an output around rating.

38 Nowhere is there such an opportunity for good return on the investment as in the boiler house, where a few dollars properly applied can perform wonders. But there are also properly designed plants that are wasteful, due to human inefficiency, and to such beliefs as that stokers are a failure, high CO_2 harmful, economizers a poor investment, superheated steam impractical, etc. It is the man who designs or operates who is at fault, rather than the equipment. The fireman is more important than the stoker and a good fireman will obtain good results from a poor stoker. But since good firemen are scarce, it is preferable to purchase equipment which is as nearly foolproof as possible.

39 When equipment is already installed but not considered satisfactory, great care should be taken to find out if it is really necessary to make a change before an expenditure is recommended.

FEATURES OF AN IDEAL BOILER INSTALLATION

40 There remains an immense field for experimental research to obtain the necessary data for boiler construction.

41 The ideal boiler installation of today should embody the following features:

a A construction permitting counterflow where flue gas will heat the feedwater which will enter the boiler at as nearly the steam temperature as load fluctuations will permit

b A combustion chamber sufficiently large and effective throughout to prevent the entrance of any flame among the tubes, thus assuring complete combustion, absence of soot, and proper cooling of the gas

c Boiler passes which will be effective throughout without dead spaces where there is no gas flow or only little, and in which the draft drop will be the minimum corresponding to gas velocity. The gas velocity will be the highest permissible by the cost of producing this velocity

d Fuel resistance will be overcome by forced draft

e The relative location of firebed and boiler heating surface is to be such that the maximum possible amount of radiant heat, consistent with complete combustion, will be transmitted to the boiler surface

f Tight boiler walls properly insulated, probably an air-jacketed furnace, the air from which will be injected over the firebed at high velocity to aid mixing effect or passed through the grate in the ordinary way

g A stoker permitting combustion with no more than 15 to 25 per cent excess air, no more than 1.2 per cent of unburned carbon in ashpit, no deposit of slag on the tubes, and these conditions maintained at low and also at reasonably high ratings

h No scale deposit in the boilers, prevented by specially constructed self-cleaning boiler, external chemical treatment of feedwater, use of distilled makeup water, filtering of the water in the boiler by recirculation, or by a combination of these methods.

DISCUSSION

WM. S. ALDRICH. In applying Fig. 3 to Mid-West coals, especially Illinois and Indiana coals, a marked difference will probably be found. Searching for this, and not having other tests of equal magnitude and number, we referred to the Steaming Tests of Coals, U. S. Bureau of Mines, Bulletin 23 (1912). Of the 111 boiler trials with Illinois coals, 75 have been grouped on the basis of the combustible consumed per hour, inasmuch as the coals from this field have approximately the same heating value on the combustible basis. The result is a remarkably uniform increment of increase of the flue-gas temperature with the capacity, within the range of the trials, from about 60 to 105 per cent steaming capacity. For such coals, therefore, the author's lines in Fig. 3 should have a decided curve upward.

What is the meaning of "very good" in this connection? In an actual case in hand, we would say 13.25 per cent CO_2 in the gases at the rear of the combustion chamber, with coal from Illinois (Franklin Co.), averaging a possible maximum of 17.5 per cent CO_2 for perfect combustion. The lowest obtained was 8.25 per cent CO_2. Here is a working range from 75.7 to 47 per cent of the maximum possible CO_2. Should the lowest value be designated as "poor" or "very poor" results? Why not make use of a percentage scale, based on the maximum possible CO_2, for perfect combustion, with the given coal?

The weight of the flue gases per pound of dry coal is equally as

important as their temperature in estimating the heat lost up the stack. This would serve equally well for a basis of comparison at various steaming capacities, except that the temperatures are more quickly and accurately determined than the CG_2, fixing the weight of flue gases by means of the excess air. For any given capacity there is quite a wide range of possible flue-gas temperatures and weights of flue gases which will give the same amount up the stack per pound of fuel. There are operating conditions under which considerable change may occur in the CO_2 without appreciably affecting the flue-gas temperature till the approach to the critical point, at and beyond which the reverse condition holds, the capacity and heat lost up the stack, per pound of dry coal, remaining constant. It is therefore necessary to know the weight of flue gases, per pound, dry coal, and the determining factor, the CO_2 per cent.

Referring to Fig. 6, inasmuch as the theoretical furnace temperature varies quite uniformly with the CO_2 for a given coal, why not use the direct relation between the CO_2 and the weight of dry gases per 10,000 B.t.u. generated?

In the case of waste-heat boilers, the hot gas relations give a simple diagram connecting the heat transfer with given capacities, through the corresponding weight of dry gases, and a consideration of their initial and flue-gas temperatures, and that of the steam.

Such analysis makes it necessary to consider carefully the points brought out by the author with regard to the heat absorption by radiation and by convection. Examining the results of trials of waste-heat boilers earlier reported to the Society by A. D. Pratt[1], we find a fairly constant average of 5 per cent of heat absorption by radiation, and more or less irrespective of the capacity. These are in a class by themselves, and might properly be styled convection steamers. For example, a waste-heat boiler, under ordinary service developed an evaporation of from $33\frac{1}{3}$ to 50 per cent of its rated horsepower. A maximum of 92 per cent capacity was obtained during the few hours of finishing heats of its heating furnace for forgings. The same boiler, with furnace arranged for direct heat, and coal firing, has easily developed 150 per cent capacity, wth no other changes than a suitable fire archway instead of the usual small checker between the heating chamber and the gasway leading therefrom to the base of the boiler.

[1] Utilization of Waste Heat for Steam Generating Purposes, A. D. Pratt, TRANS., v. 38, 1916, p. 599.

R. SANFORD RILEY. The writer wishes to express his appreciation and he believes the appreciation of many others interested in the variety of apparatus with which the author has dealt, for the care with which the paper has been prepared. Nothing sounder has been expressed before the Society than the author's ideas on differences in conditions of operation. The series of tests quoted show that increase in efficiency and capacity are very frequently possible and the author's analysis is very much to the point.

D. S. JACOBUS. I wish to add a word to what the author has said about the necessity of considering other factors than the efficiency of the boilers. Much of the saving possible in an industrial plant may come through other improvements than in the boilers, such as in the manner of heating the feedwater, the returning of the condensed steam with its contained heat from various processes to the boilers, and the heating of the plant in cold weather.

DAVID MOFFAT MYERS. The writer once effected a fuel saving of seven per cent without making any recommendations or giving any directions. He had been called to the plant to make an investigation. The firemen in the boiler room saw him about the plant and learned the object of his visit. After a few days the superintendent told the writer that the daily coal consumption had dropped off seven per cent. This simple incident serves to illustrate the importance of the psychology factor to the fuel problem in power plants.

The fireman may know how to produce results but unless he has some incentive to produce them the efficiency will be low. The problem involves a proper handling of the human, as well as the mechanical, factors as these two essential parts are unavoidably and continually bound together.

C. H. SMOOT. The paper states that the efficiencies of boilers were raised to 70 per cent without making changes in the apparatus. Someone is responsible for the inefficient operation. Is it the manufacturer of the boiler or the manufacturer of the stoker? Is it a question of the care given the apparatus or of the education of the fireman? If without change in apparatus the efficiency of a boiler can be raised from 50 to 70 per cent by giving the minor attention described by the author it seems as though some organization is necessary which will see that this is done with all boilers. By fail-

ing to take advantage of these conditions we are losing sight of one of the greatest objects which was contemplated in the organization of this Society.

C. Harold Berry. The author is undoubtedly referring to small plants because many of the things he says do not apply to large central stations. It should be pointed out in connection with Table 3 that while this was a great increase in efficiency in the three tests, nevertheless the percentage of rating was notably different, increasing from 102 per cent to 140 and finally to 185 per cent. This will not account for the entire improvement but must be considered in the interpretation of the results.

In Par. 34 the author's statement that a large majority of plants operate too many boilers does not apply to large central stations. His statement in Par. 36 that good boilers well installed should be operated at 150 to 200 per cent of rating is in the writer's opinion correct. If there is a choice between operating at lower rating and banking it is the writer's experience that it is better to operate at lower rating. The overall economy for twenty-four-hour load conditions in central stations is better than if some boilers are banked.

The curve of Fig. 10 which indicates the efficiency of an oil-fired boiler decreasing as the rating increases is contrary to the condition which prevails in coal-fired boilers where the efficiency rises to a maximum at about 140 to 150 per cent of rating.

The author replied to Mr. Berry and said that the tests reported in Table 3 were run to obtain in each case the maximum capacity possible and that the improvement may increase the capacity as well as the efficiency.

J. M. Spitzglass. The writer was once called in to settle an argument between an architect and a heating contractor in a factory where the boilers were not sufficient to deliver the steam to heat the building. He started a test. A regular fireman was employed in the building, one who knew how to handle a poker and a shovel. It was really a pity to watch the firemen doing the hard work necessary to fire the boilers. Still the building was cold and the pressure was low. To help matters out, the heating contractor called on the boiler people. They came with their own fireman to prove the capacity of the boilers. It appeared to the writer that the new fireman did not work hard at all, yet normal pressure was soon reached, as was the heating capacity of the boiler, and what was more, the

amount of coal passed through the fire door was very greatly diminished.

What would be expected under those circumstances? Hire a good fireman to attend to that job. That is easier said than done — it is easier to speak about a good fireman than to find him.

Mr. Azbe covered the subject of boiler-plant efficiency very thoroughly. The last paragraph is, the writer believes, the keynote to boiler-room efficiency, and consequently means the keynote of Mr. Azbe's paper.

In Par. 14 Mr. Azbe says — "As to fuel oil, it should be made an offense against the law to burn crude oil under boiler furnaces." I would say it should be considered an offense against the law to employ a fireman who does not know how to fire the boiler efficiently.

In this campaign we are having now to increase efficiency and eliminate waste, the Society could well afford to use its influence to inaugurate a system of good firemanship by making the work interesting to good people, better men, and by passing a law that the fireman should be required to have a license, that he knows how to do this work, the same as his more fortunate brother in the engine room.

THE AUTHOR. Mr. Berry is not entirely right in assuming that the author refers to small plants only; most of the observations upon which the paper was based were made in large industrial plants and the fundamental information given can as well be applied to central station as to other boiler plants. Of course the larger central stations pay a great deal of attention to fuel economy, but we should not let that blind us, and assume, as we have the tendency to do, that large central stations' economy received is representative of what is received in other industries. There is far too much written about good plants and far too little about poor ones which distorts our opinion of the actual conditions.

As to Table 3, the author wishes again to state that in each of the three tests, maximum capacity was striven for. During the first test only 102 per cent of rating was obtained due to poor economy and in later tests capacity was increased because economy was increased. Capacity and economy are interdependent but such statements as "efficiency rises to a maximum at about 140 to 150 per cent of rating" as made by Mr. Berry should no more be perpetuated. The maximum point of the efficiency curve depends upon a great many conditions, many of them being given in this paper.

Some of the points taken by Mr. Aldrich are excellent, but in defense of Fig. 3, the author must say that up to now no measure of what is good or poor flue-gas temperature exists. We know what is good or poor CO_2 but the engineer was handicapped when he attained high CO_2 in determining other directions in which he still further could improve the efficiency; his flue-gas temperature may have been 600 deg. fahr., but what is that, good or bad? It may have been both, therefore, some two hundred tests were analyzed for the purpose of constructing this chart and it should well serve as a comparative measure judging one's boiler performance with another's.

No. 1813

BOILER AND FURNACE ECONOMY

By D. S. Jacobus, New York, N. Y.
Member of the Society

That it is possible to effect material savings in power plants is evident, according to the author, when it is considered that fifty or even a hundred per cent more fuel is burned in some cases than would be required by plants of the best economy. It does not follow, however, that it would pay to replace all plants of low efficiency by those having a higher efficiency. There are many factors which must be taken into consideration before deciding that a new investment is warranted, and each case must be considered individually in order to determine whether or not it will pay to replace an existing plant by one of more economical type, or to add apparatus to increase the efficiency.

The present paper accordingly deals with such questions as when it will pay to use economizers in a new plant; the most economical rating at which to operate a boiler; limitations imposed on efficiency by inability of furnace brickwork to withstand temperatures available with many classes of fuel; furnace volume and length of flame travel; where air heaters are of a special advantage; importance of employing properly trained men for boiler operation; information afforded by flue-gas analyses, etc., etc.

The author points out in closing that furnace and boiler design must be coördinated in order to secure the best results. It is impossible to separate the boiler efficiency from the efficiency of the stoker and furnace in such a way that the value obtained for the former will depend solely on the construction of the boiler and not be influenced through the construction of the stoker and furnace. The stoker and furnace efficiency is influenced by the delayed or secondary combustion between the boiler tubes and this cannot be included in any analysis, neither can the loss due to excess air be correctly divided between the boiler and furnace.

THERE has been a steady improvement in the economy of power plants, on account of the increase in fuel costs and to keep pace with the advance in the art. Particular attention is now being given to savings in view of the economic struggle which appears to be before us. That material savings can be effected in power plants is plain when it is considered that 50 or even 100 per cent more fuel is burned in some cases than would be required by plants of the best economy. It does not follow, however, that it would

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pay to replace all plants of low efficiency by those having a higher efficiency. There are many factors which must be taken into account before deciding that a new investment is warranted and each case must be considered by itself in order to determine whether or not it will pay to replace an existing plant by one of more economical type, or to add apparatus to increase the efficiency. Adding an economizer will ordinarily add to the thermal efficiency; the same applies to an air heater, but on considering all features it may be found that the character of the load, the class of operation, and the fuel conditions are such that it will not pay to install an economizer or an air heater.

WHEN IT WILL PAY TO USE ECONOMIZERS IN A NEW PLANT

2 In determining whether it will pay to use economizers in a new plant the problem should not be approached by comparing the efficiency of a boiler with that of the same boiler with an economizer added to it. The proper way is to compare the results to be expected from boilers best suited for the service without the addition of economizers, with the results to be expected from the best combination of boilers and economizers. It will sometimes be found for certain load conditions that a larger boiler properly designed and baffled will give better commercial returns, all features considered, than a smaller boiler with an economizer, or for that matter, than any boiler that can be selected with an economizer. In the case of an old plant, the question of whether or not it will pay to add economizers to the boiler is a simpler one to answer than for a new plant, as the added efficiency due to the economizers may be readily computed on the basis of the known flue-gas temperatures, gas weights, etc.; but here again the load conditions must be taken into account, as well as the facts that there will be an increased cost of upkeep and added complication through the necessity of installing an induced-draft apparatus. For peak-load service where the peaks are of short duration, say an hour or so a day, it does not pay in most cases to install economizers on all of the boilers. A good arrangement may be secured for some classes of service by adding economizers to some of the boilers and operating them at a more nearly uniform load than the rest of the boilers, the boilers that have no economizers being held in reserve and cut in during the peak-load periods. Another case where it does not pay to apply economizers is where boilers are used for stand-by service. Where

the operation is more or less continuous it pays in many cases to use economizers. The cost of fuel is a governing factor. The increase in fuel costs during the last few years has caused economizers to be used more generally than formerly and we are approaching more closely to European practice in the number of boilers fitted with economizers.

INCREASING EFFICIENCY BY ADDING TO HEIGHT OF BOILER

3 In designing a boiler for use without an economizer additional efficiency may be secured by adding to the height of the boiler. Adding to the height of the boiler so as to increase it from, say, 14 tubes high to 20 tubes high, will result in a considerable increase in efficiency with but little increase in the draft loss. In many instances the higher boilers may be operated with natural draft at the desired capacity, thereby eliminating the added complication of the induced-draft apparatus that would be required for economizers. Of course, it would be impossible to operate a high boiler of the sort with natural draft at as high a percentage of rating as a lower boiler, as the flue-gas temperature at a given rating would be considerably lower than with the lower boiler, and this would reduce the amount of draft available from the stack.

4 Wrought-steel economizers are coming into use to a greater extent than formerly. We have learned how to prevent undue interior corrosion in a wrought-steel economizer by reducing the oxygen content of the feedwater with as low a temperature of feedwater as can be employed in a cast-iron economizer. To avoid exterior corrosion through the condensation of moisture from the flue gases, the temperature of the feedwater to the economizer must be kept above a temperature of about 120 deg. fahr., and for most work 140 deg. fahr. is preferable, as this allows for some leeway in case the water is fed intermittently. There is more difficulty through exterior corrosion in intermittent service than where the service is more or less continuous, as soot which collects on the outside of the economizer is apt to become moist during the time that the economizers are down or are being started up, and the presence of moisture makes the soot from certain fuels highly corrosive. Coals having a high sulphur content give the most trouble through exterior corrosion, and with such coals the temperature at which the water is fed to the economizers should be somewhat higher than for coals having but little sulphur.

AN INDIVIDUAL ECONOMIZER FOR EACH BOILER

5 It is becoming more general practice to use an individual economizer for each boiler and not to install connections for by-passing the flue gases around the economizer. A by-pass connection as a rule allows some leakage of the hot gases around the economizer and causes a continual loss of efficiency. Omitting the by-pass connections is also advantageous in simplifying the operation and lessening the chance of trouble with the economizers.

6 The efficiency obtainable from the boilers is not the only factor that governs the efficiency of the plant as a whole, especially under variable load conditions. The variation in the load must naturally be taken care of either by operating the boilers at widely varying capacities or by carrying banked boilers which are cut in at the time of the peak loads. It can be readily appreciated that should the boilers at a plant be limited to a capacity of 200 per cent of rating, a greater number of boilers would have to be in continuous operation or a greater number of boilers would have to be held in reserve for cutting in during the peak-load interval than were it possible to operate the boilers at the peak-load intervals at 300 or 400 per cent of rating. By arranging to operate the boilers at 300 to 400 per cent of rating they would ordinarily not give as high an efficiency at the lower ratings as the boilers having stokers especially fitted for operating at the lower ratings, but when it comes to the efficiency of a plant as a whole, a better efficiency would ordinarily be secured by the use of boilers capable of being operated at the higher ratings even though these boilers might be, say, 2 per cent less efficient at a given rating than the boilers having a more limited capacity. The obtainable capacity is therefore an important factor, especially when the load is variable. In the event of extreme variations which come at rare intervals it can be readily seen that the ability to carry the load far offsets any loss in economy when running at the higher load, and it is these elements that have caused the capacity at which boilers are run for peak-load service to be advanced to 300 or even 500 per cent of rating. High capacities of the sort should not be attempted, of course, without the very best of feedwater and the highest class of attendance.

THE MOST ECONOMICAL RATING AT WHICH TO OPERATE A BOILER

7 A question often arises as to what is the most economical capacity or rating at which a boiler should be run. This cannot

be answered in a general way as it depends on the load conditions as well as the class and cost of the fuel. The economical rating is naturally influenced by the presence or absence of economizers and the service to which the boilers are put. Where there are short peak-load periods the greatest commercial economy is usually secured by operating the boilers to as high a capacity as can be secured during these periods. Each case must be considered by itself and the fuel economy worked out for the actual conditions of service. The effect of the additional cost of upkeep must naturally be considered. The general practice in this country is to drive boilers at a higher rating than is done in European practice, and the stress of war conditions undoubtedly led to a number of plants being run beyond the point of the best commercial efficiency. The fact remains, however, that boilers are being run at higher and higher capacities in our modern power plants and the tendency seems to be upward rather than downward.

8 With a uniform load, or a practically uniform load, the problem of determining the most economical capacity at which to drive boilers is a simpler one than with a fluctuating load, but even here the point of best economy from a thermal standpoint is considerably lower than the point at which the boilers should be operated to secure the best commercial economy.

LIMITATIONS IMPOSED ON EFFICIENCY BY FURNACE BRICKWORK

9 For securing the highest efficiency, the furnace temperature should be the maximum that can be maintained, and combustion should be completed within the furnace chamber. The furnace brickwork employed today will fail if saturated with heat at the full temperature available with many classes of fuel. In furnace design, therefore, some efficiency must be sacrificed in most cases in order to maintain the furnace brickwork and keep the cost of repairs within a reasonable figure. Should it be possible to secure for furnace construction refractories that could be brought up to the full available temperature, and that would withstand the effect of erosion of the gases and be free from spalling, the art of furnace design would be considerably changed. As the art stands today, most of the thickness of the brickwork must be cooled to a temperature below, say, 2200 deg. fahr., in order that the bricks or tiles will carry the load imposed on them. To accomplish this a sufficient amount of the heat absorbed by the brickwork must be

dissipated either through radiation or convection. Where furnaces are operated under a suction which causes the cool air to be drawn inward through the brickwork there is a cooling effect which serves to prevent overheating, whereas if they are operated under a pressure there is a tendency for the hot gases to leach outward and overheat the brickwork. The difference between running with a slight suction of, say, 0.1 in. water column and a plus pressure within the furnace may make all the difference between successful operation and having a lot of trouble. Should it be possible to operate with a plus pressure and maintain the brickwork irrespective of the temperature to which it would be raised, more economical results could be secured in most cases through the elevation of the furnace temperature. The exception would be where the elevation of temperature would interfere with the combustion; for example, with some kinds of coal too high a temperature might fuse the ash and clinker and cause the coal to mat down on the grate so as to increase greatly the air resistance through the fuel bed and reduce the available capacity. Again, should it be possible to bring the brickwork up to the full furnace temperature, arches could be used for certain purposes in furnace design where they cannot be used at present.

10 Brickwork will fail by plastic deformation before it reaches the melting point. The greater the load carried by the brick, the more apt it is to fail. Fireclay brick of the best quality ordinarily obtainable begin to show plastic deformation under a load of 20 lb. per sq. in. at from 2200 to 2400 deg. fahr. Reduction to a load of 10 lb. per sq. in. will increase the permissible temperatures about 200 deg. fahr. As furnace temperatures considerably higher than this exist with certain grades of fuel and stoker practice, say, 2700 to 3000 deg. fahr. as a limit, it is apparent that the necessity of maintaining the brickwork below the temperature of the furnace is a vital one. First-class clay brick have a fusion point slightly above 3100 deg. fahr. and they yield through plastic deformation long before they fuse. This emphasizes the importance of conducting proper tests for plastic deformation.

11 In ordinary furnace design the walls and arches are heated on one side only in order that the brickwork may be maintained at a lower temperature than the full temperature of the furnace. Arches will give trouble if heated on both sides which would not give trouble if subjected to the same amount of heat on one side only. This sometimes makes necessary the use of double arches,

one directly above the other, with a small amount of air admitted between them for ventilating and cooling. Where battery walls are heated on both sides they must be made thick enough to conduct the heat from the hotter to the cooler parts of the walls to a sufficient extent to keep the wall at a temperature below that which will cause trouble. A wall of a given thickness that would give good service as a battery wall might fail for the same fuel and combustion conditions if used for supporting combustion arches within the furnace, as the heat would not be conducted away from the supporting wall to the extent that it would in the case of a battery wall. A supporting wall of the sort should be used between combustion arches only in cases where a low grade of fuel is burned which does not result in high temperatures, or it should be ventilated.

POINTS TO BE CONSIDERED IN DESIGNING AN EFFICIENT FURNACE

12 To consume the combustible gases within the furnace chamber there should be a proper length of flame travel before the gases strike the tubes, and a sufficient furnace volume. Furnace volume and length of flame travel are not, however, the only elements that must be considered in designing an efficient furnace, as there must be a mingling action within the furnace to cause any unconsumed combustible gases to reach the excess air. It is possible in some forms of furnaces for a lane of gases containing excess air to pass entirely through the boiler to the uptake. To secure the best efficiency the furnace should be of such a form as to prevent a laning action of the sort. Gases containing excess air will ordinarily mingle more readily with the hotter gases containing unconsumed combustible matter if they are made to travel over and above the hotter gases within the furnace, as this leads to an eddying action through the greater density of the gases containing the excess air. Where two lines of stokers are used with the dumping plates at the middle line of the furnace, the furnace should be made high enough to allow the lean gases which come at the middle of the furnace to form eddying currents with the hotter gases and thereby to search out the combustible matter in the hotter gases. In some cases a swirling action can be brought about which increases the turbulence and adds to the efficiency of the combustion. In a furnace of this sort the stokers should be set so that the dumping plates come at right angles to the horizontal projection of the tubes, because if

set at 90 deg. to this position the quality of the gases from side to side of the boiler would not be uniform.

13 In the burning of gaseous, liquid, and powdered fuel a laning action, or an irregular mingling of the fuel and air, may lead to pulsations within the setting, which may become so severe as to limit the capacity at which the boiler can be operated. The remedy consists in obtaining a proper distribution of the air and fuel. The velocity at which the air is admitted is an important element and there is usually less trouble experienced where a forced blast is used than under natural-draft conditions.

14 There has been a gradual increase in furnace volumes. In the burning of blast-furnace gas we now furnish at least 2 cu. ft. of furnace volume per rated boiler horsepower, which is over double that used in the older practice. For some coal-fired boilers fitted with forced-blast chain-grate stokers we are supplying about 4 cu. ft. per rated horsepower. The gas-fired boilers are operated at about 200 per cent of rating as a maximum and the coal-fired boilers at over 300 per cent, so that in each case we are supplying about 1 cu. ft. of furnace volume per maximum developed horsepower. In the case of powdered-fuel furnaces the furnace volume of recent installations is about 2 cu. ft. per developed horsepower. These figures are for the larger furnaces used in stationary practice, and in some instances with smaller furnaces a higher capacity may be developed per cubic foot of furnace volume. It is seldom, however, that over 2 hp. is developed in stationary practice per cubic foot of furnace volume. In marine practice, where the furnaces are made of the smallest size that will give good results, as high as 5 to 6 hp. may be developed in both coal- and oil-fired boilers per cubic foot of furnace volume. High capacities of the sort with relatively small furnaces can only be secured with the cleanest of feedwater, and it is certainly advantageous to use the larger furnaces in stationary practice, as by so doing it is easier to operate without an undue amount of tube troubles, the trouble through the collection of slag on the exterior of the tubes will be diminished, and the combustible gases can be more thoroughly consumed within the furnace chamber.

15 The effect of the absorption of radiant heat on the boiler tubes should be considered in designing a furnace. With certain fuels such as blast-furnace gas, wet wood or bagasse, where the highest attainable temperature can be carried by the brickwork, it is best to absorb but little radiant heat in order to maintain a

high furnace temperature. With the stronger fuels it is necessary to absorb a considerable amount of the radiant heat in order to prevent the collapse of the brickwork or an undue amount of deterioration. Few realize how great a proportion of the heat may be absorbed by a boiler as radiant heat. In underfeed-stoker practice, with the baffle arrangement ordinarily offered with Babcock & Wilcox boilers, the absorption of radiant heat with the boiler operated at its normal rated capacity is over 60 per cent of the total heat absorption. Exposing more or less of the boiler surface to the direct radiant heat of the fire with the stronger fuels has a comparatively small influence on the efficiency, as exposing a greater amount of surface reduces the furnace temperature which offsets some of the gain, and as any increase in the heat absorbed through direct radiation is also counterbalanced to an extent by the diminution of the amount of heat absorbed through conduction. Ordinarily, the higher the furnace temperature, the higher the efficiency. Higher furnace temperatures, however, lead to increased cost of brickwork maintenance, especially when the boilers are operated at high ratings, and for a strong fuel it usually pays to expose a considerable proportion of the heating surface of the boiler to the direct action of the radiant heat.

16 A furnace to give economical results at high ratings involves special problems in caring for the expansion of the brickwork and especial care in the construction of buckstays for holding the brickwork in alignment. The general tendency in large furnace walls which are highly heated is to bulge inward toward the fire, and unless some means is provided to prevent this they may collapse. In our present practice we use horizontal rows of bonding tile which are held by cast-iron bulb pieces attached to the buckstays in such a way that the wall can expand in any direction in a generally vertical plane, whereas it is prevented from either bulging inward or outward. Another means that may be employed to prevent the walls bulging inward toward the fire is to build them with a camber with a vertical axis, that is, curved slightly, with the concave side next the furnace. Still another way is to incline the walls outwardly.

REMEDIES FOR CERTAIN OPERATING DIFFICULTIES

17 In operating at higher ratings difficulties are encountered with some grades of fuel through slag adhering to the boiler tubes

and restricting or closing up the passageways for the flow of the gases. The difficulty through slag can be reduced by providing a relatively large area for the flow of the gases on entering the spaces between the tubes and by furnishing access doors through which the slag can be detached from the tubes. An air or steam lance, which is used both for cooling the slag and as a rod for striking it to remove any portion that may adhere to the tubes, forms an efficiency tool for use in connection with the access doors. The difficulty through slag is essentially a stoker operating difficulty. Carrying a thick fire may result in but little slagging action, whereas this may cause trouble with the stoker itself through clinker. Thinner fires may remedy the stoker difficulty but may result in an excessive amount of slag being deposited on the boiler tubes. With certain grades of coal there will be some accumulation of slag on the boiler tubes with the stokers most carefully operated, and it is advantageous in such cases to provide access doors for removing the slag. Steam-jet blowers serve to remove or to reduce the amount of slag which will collect on the tubes, but are not a complete remedy in all cases.

18 Trouble through clinker at the sides and front walls of the furnace where underfeed stokers are used is often reduced through admitting air under a blast pressure through a number of openings through the walls at the sides of the fuel bed. The air flows in a layer against the inner face of the wall and prevents the clinker adhering to the walls. It also serves to partially cool the wall and reduce the erosion of the wall. This method was applied to one of our boilers at Dayton, Ohio, in 1912, and has since been extensively used. A difficulty that arises in some cases is the formation of a clinker which will cause the air admitted under a blast pressure to produce a jet or torch-like action on the furnace walls. This is clearly a stoker operating difficulty. In some cases a torch-like effect of the sort will result in fusing the brick so as to melt out or erode away a part of the wall. The stoker tuyeres may also cause jets of air to be projected against the wall so as to erode away certain parts of the wall.

WHERE AIR HEATERS ARE OF ESPECIAL ADVANTAGE

19 Air heaters are of especial advantage where low grades of fuel are burned and it is desirable to increase the furnace temperature. In such cases the increase of efficiency of the boiler or

boiler and economizer as a whole may be greater than that corresponding to the amount of heat imparted to the air heaters, as the heated air may increase the efficiency of the combustion to such an extent as to result in a greater gain in the heat absorption than that corresponding to the heat imparted to the air. Of course, the gain through the use of the air heater would be measured by the efficiency of the plant when run with and without the air heater but the difference obtained in this way might be greater than that corresponding to the amount of heat absorbed by the air heater.

20 In the case of certain fuels an air heater may not be desirable on account of increasing the trouble with clinkers in the furnace, and also on account of increasing the upkeep of the brickwork. As higher and higher efficiencies are sought, air heaters will undoubtedly be used to a greater extent the same as in the case of economizers. In common with economizers, there may be trouble with air heaters through a sweating action of the gases in depositing moisture and acid on the colder portions, and all of these features should be considered before deciding to use an air heater.

IMPORTANCE OF EMPLOYING WELL-TRAINED MEN FOR OPERATION OF BOILERS

21 Irrespective of how well a boiler plant may be designed, it will not give the best results unless it is properly operated. The man in charge of the boiler room is the "man behind the gun" when it comes to securing the best obtainable efficiency. The old idea that a man should graduate from the boiler room to the engine room is a mistake, as a good man in the boiler room can increase the efficiency through good operation far more than one in the engine room. Happily more attention is now being paid to this feature, with correspondingly good results.

22 All of those who have had much to do with increasing the efficiency of power plants know that often the simplest requirements of good practice are disregarded in the operation of boilers, the boilers often being dirty and run with leaky baffles and settings. Even where the boilers and settings are kept in good condition there may be loss through improper firing. The fireman is often blamed as a factor of fuel waste. The difficulty is seldom with the fireman, but is in having nothing to guide the fireman except what might be called his intuition. It is certainly a mistake not to provide proper instruments. This is well appreciated in the larger

and best-run power plants, but there is often a lack of instruments that makes it impossible to operate the plant to the best advantage. There is no reason why operating results should not be as good for a given set of conditions as test results, and this can only be accomplished through installing high-grade indicating and recording apparatus for the guidance of the fireman and operators. The plants of today are of a more complex character than in the past and the success in operating these plants depends in a great measure on the use of the proper instruments and a proper training of the men.

INFORMATION AFFORDED BY FLUE-GAS ANALYSES

23 In the operation of a plant much is shown by the flue-gas analyses. The flue-gas analyses are not all that govern the efficiency, as much depends on whether or not there is delayed combustion between the tubes of the boiler. It is all wrong to assume that if two boilers are operated with the same average flue-gas analyses and percentage of carbon in the ash that the furnace efficiency is the same in each, as actually there may be a wide difference due to the fact that all of the combustible gases may be consumed in the furnace in one case and not in the other.

24 A small amount of CO is often present when a high percentage of CO_2 is carried. An analysis with coal firing which indicates 13 per cent of CO_2 and no CO is about equivalent from a heat-loss standpoint, where the flue gases leave a boiler and economizer at 300 deg. fahr., to one which gives 15 per cent CO_2 and $\frac{1}{4}$ percent of CO. Again, carrying a high percentage of CO_2 sometimes leads to secondary or delayed combustion between the boiler tubes, thereby involving an additional loss through increasing the temperature of the flue gases. It is important, therefore, in boiler-room practice to make accurate analyses to determine whether or not CO is present in the flue gases and also to avoid too great an amount of delayed combustion between the boiler tubes.

25 Automatic systems for combustion control are often used to advantage. The supply of fuel is regulated in some cases to meet the demand for steam as well as the amount of air supplied for combustion. Systems where the amount of fuel is automatically regulated are in more general use in oil burning than for coal. In most cases the air supply is regulated to correspond to the fuel burned and to give a proper amount of draft within the furnace and the fuel supply is hand-adjusted. Such systems quickly regu-

late the air and draft, and in some cases the fuel as well, to meet changing load conditions, and are especially useful where there is a variable load.

26 High-class instruments and apparatus for guiding the firemen or controlling the combustion will not of themselves insure good results. Many such devices that would give highly satisfactory results in the hands of a capable man that believes in them would be a failure in the hands of one less capable or one prejudiced against them. Here again all depends on the man in charge, and as the art advances and more complicated apparatus is used in obtaining higher efficiencies, more competent men are required.

NECESSITY OF COÖRDINATING BOILER AND FURNACE DESIGN TO SECURE BEST RESULTS

27 Furnace and boiler design must be coördinated in order to secure the best results. It is impossible to separate the boiler efficiency from the efficiency of the stoker and furnace in such a way that the value obtained for the efficiency of the boiler will depend solely on the construction of the boiler and not be influenced through the construction of the stoker and furnace. Many have worked on this problem and the ground has probably been gone over more thoroughly than any other feature in boiler-testing codes. The stoker and furnace efficiency is influenced by the delayed or secondary combustion between the boiler tubes and this element cannot be included in any analysis, neither can the loss due to excess air be correctly divided between the boiler and furnace. The element of delayed combustion may influence the overall boiler and furnace efficiency, say, 5 or 10 per cent with the same flue-gas analyses. In most proposed methods the stoker and furnace efficiency is based on the analyses of the flue gases and on the amount of unconsumed carbon in the ash, and it can be readily seen that such a basis cannot give reliable results. Should it be possible to express accurately the furnace and stoker efficiency, a purchaser should obtain the same overall economy from his boilers with any arrangement of stokers and furnaces having a given efficiency. There is no method whereby this result could be insured and whereby the purchaser would be protected should he assume that the boiler and furnace efficiency would be an exact measure of the relative advantages of using one or another of the various stokers and furnaces in connection with his boilers.

DISCUSSION

ALEX D. BAILEY. Referring to Par. 19, the statement is made that air heaters will be of special advantage where low-grade fuels are burned. While this might be true as stated, it is equally true, particularly in the case of coals, that on account of the sulphur content of these same fuels, air heaters would operate at the greatest disadvantage.

In the first place the rate of heat transfer for this type of apparatus is low, so that large surfaces are necessary, with consequently restricted passages for air and flue gas. Further, the temperatures at which these would operate are so low that the surfaces exposed to the flue gas would soon be coated with a pasty acid-bearing soot which it would be almost impossible to remove by ordinary means, and which, at the temperatures used, would corrode the surfaces very rapidly.

It seems that the nature of the fuel itself, should be a preliminary consideration in any discussion of air heaters.

So far as the personnel of boiler plants is concerned, probably the greatest need for education and missionary work is with managers and chief engineers who do not appreciate the importance of this part of power-plant operation.

In the past it was customary to promote men from positions as unskilled laborers, coal shovellers, boiler cleaners and the like to that of firemen, but this day is past and the man who can come through this course of promotion and make a good fireman in the proper sense is the exception, as he generally lacks the fundamental education necessary.

In order to attract the proper men for this work, boiler-room conditions must be improved, as well as working conditions, and it must be realized that the job must have attached to it the standing and importance and pay which it deserves and commands.

Automatic systems for combustion control will never take the place of brains, and while such systems may function fairly successfully if left alone, their ultimate success would be improved by proper supervision and care.

Instruments likewise, do not insure proper boiler operation unless there are brains and ability to make use of them and their indications. If the fireman does not understand and cannot interpret their readings, it would be as well to give him a set of books on the theory of combustion and expect him to digest them.

To attract the right kind of men the conditions in the boiler room must be improved, then with proper equipment provided, men with education and training can get the desired results.

E. L. Hopping. The writer believes that Dr. Jacobus has covered the subject of modern boiler and furnace practice very thoroughly. Many of the points brought out in the paper lead to the conclusion that large furnaces are essential if the best economies are to be effected, especially where boilers are operated at high ratings.

In designing furnaces for 1500-hp. Stirling boilers with integral economizers, at the Delaware station of the Philadelphia Electric Company, two considerations were kept in mind:

a That the furnace be made sufficiently large to allow for efficient operation at high ratings without detriment to the furnace walls or to the stoker parts

b That in case it was found desirable, after installation of the boiler, to change to some other type of fuel burning, the cost of such a change would not be prohibitive and that conditions would be such as to produce the highest economies.

The furnace as finally designed and built has a volume of 4.8 cu. ft. per rated boiler horsepower, or 1.45 cu. ft. per maximum developed horsepower, which is a rather high ratio for stoker-fired boilers. These boilers were designed to operate at maximum capacities of 330 per cent of normal rating. The furnace as designed is shown in Fig. 1. Fig. 2 shows how the furnace may be adapted to the burning of oil or pulverized coal.

The results obtained in operation of these boilers have been very satisfactory and one condition which was not looked for has been secured. This is the practical elimination of the cinder nuisance at this plant. In most of the large stations, having stoker-fired furnaces, the cinder nuisance has been a very serious problem and efforts to correct this trouble have, in a good many cases, been very costly. From the results obtained in this direction, the writer feels that the solution of future cinder problems is to be found in the correct design of the boiler furnaces.

In boiler operation, especially at the higher ratings, there are large quantities of small particles of fuel driven off from the fuel

bed and carried with the gases up into and through the tubes. Where the distance between the fuel bed and the boiler tubes is small, these particles, while incandescent, are thrown against the cooler surface of the tube and chilled. These particles being very light pass

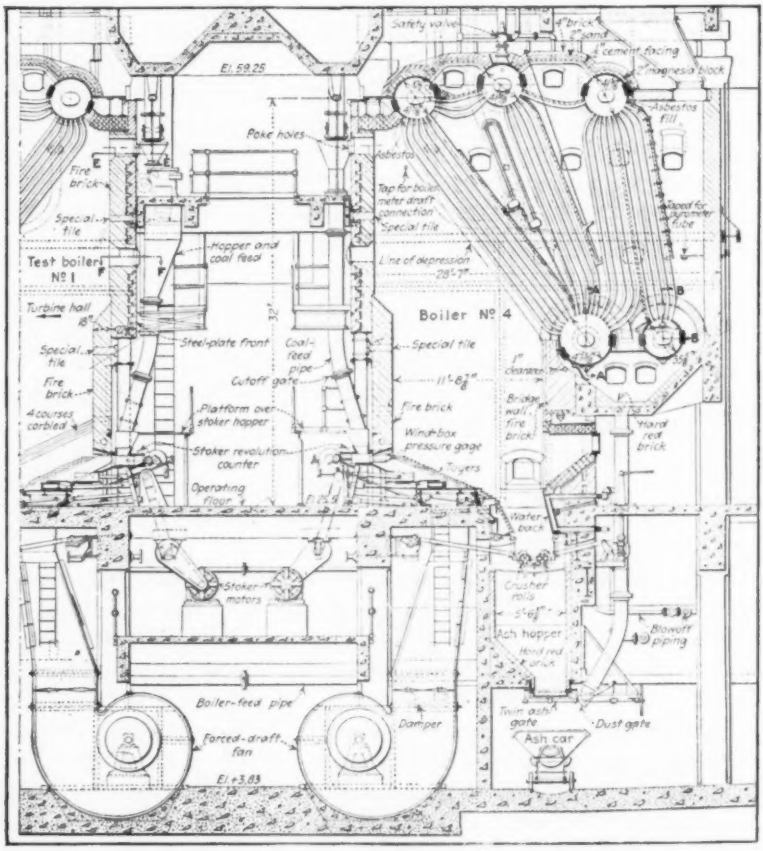


FIG. 1 CROSS-SECTION OF BOILERS AND FURNACES, DELAWARE STATION, PHILADELPHIA ELECTRIC CO.

along with the gases out through the flues and stacks and are deposited on the buildings and grounds in the vicinity of the plant; the lighter particles being carried very often to considerable distances. In this particular installation, the distance from the fuel bed is great enough to allow these particles to be almost entirely consumed before reaching the tube surface. Therefore, there is but

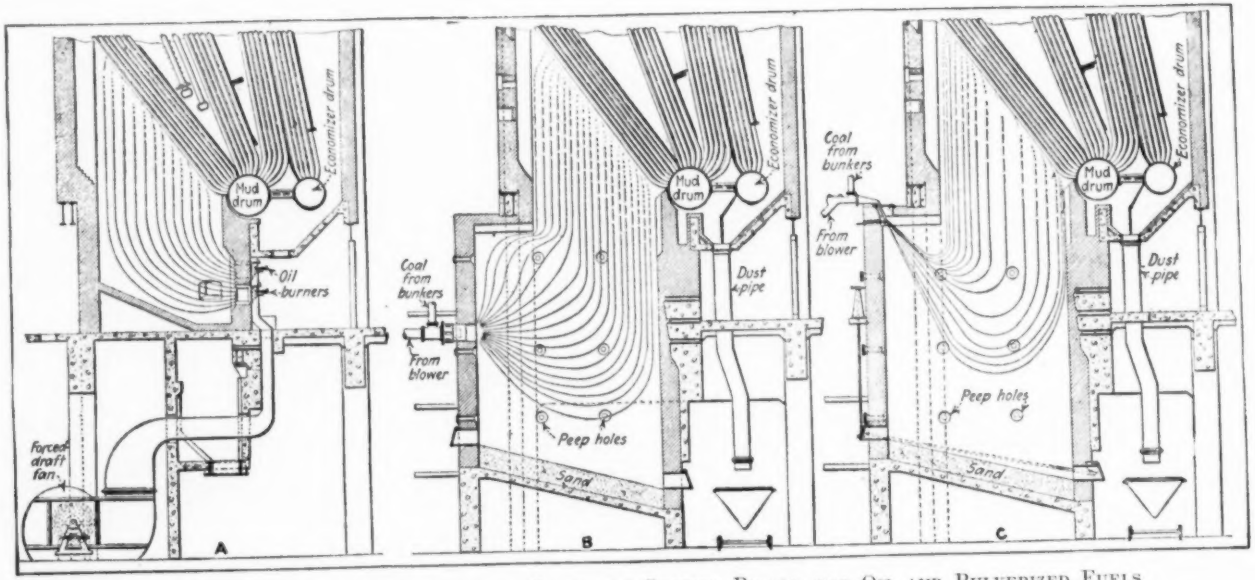


FIG. 2 PROPOSED REARRANGEMENT OF DELAWARE STATION BOILER FOR OIL AND PULVERIZED FUELS

a very small quantity of cinders to be disposed of. The few cinders that are formed are mostly caught in pockets at the lower ends of the boiler passes. After a year's operation, the roof and surrounding property at the Delaware station are still entirely free of cinders.

For any boiler plant located close to residential or manufacturing centers, where cinders are objectionable, it would seem that the question of large areas in furnaces should be very carefully gone into and the benefits obtained from such large furnaces would result not only in high operating economies but in a considerable saving in the installation of cinder catchers, or perhaps in the payment of lawsuits which might be brought against the operating companies, because of the cinder nuisance.

In Pars. 19 and 20, special reference has been made to air heaters for use in increasing the temperature of the air for combustion. The writer would like to ask the author if he can give approximate temperatures to which it would be safe to preheat air for combustion with some of the principal grades of fuel and assuming that the furnace is constructed of the best grade firebrick.

W. F. M. Goss. The paper constitutes a concise statement of significant facts. In its historical aspects its presentation is a wonderful tribute to the work of this Society. There was a time when papers dealing with boiler efficiency were replete with descriptions of methods of testing; when its membership was tremendously interested in the details of such work. Here is evidence that we have passed that period and have reached a time when we are prepared to accept concise statements with the simple assurance that they are based on knowledge. The formation of our Society was practically coincident with the beginning of boiler testing, and its proceedings constitute a record of progress in this field, with this paper of Professor Jacobus as the last word to date.

The paper also is a monument to the industry and scientific skill of its author. He has long had a part in the development of this field of engineering research. Much that is common knowledge today has come through the painstaking processes which he has inspired and promoted. Underlying this paper therefore are activities incident to his fruitful life.

Edwin B. Ricketts. As Dr. Jacobus has stated, the melting temperature of a good grade of firebrick is considerably above that usually obtained in boiler furnaces. This temperature, therefore,

has for us only an academic interest. Furnace brick work, however, seriously limits furnace temperature and, consequently, furnace efficiency by its tendency to deform under load and to form a eutectic with ash at temperatures as low as 2200 deg. fahr. Our problem is, therefore, to maintain high furnace temperatures and at the same time keep the brick-work temperature at a point below that at which it will deform under load or be attacked by fused coal ash. How is this to be accomplished?

The deformation temperature can be considerably increased by building the furnace narrower at the bottom than at the top, thus relieving the inner lining of some of its superimposed load. In many installations the brick-work temperature is reduced relative to that of the furnace by forcing currents of air or steam parallel to the walls; both of these remedies are, however, susceptible to application only over a limited range of temperatures.

In metallurgical furnaces where much higher temperatures than are obtained in boiler work have been common practice for many years, this difficulty has been met by imbedding pipes through which water is circulated in the brick work or by using thin walls, against the outside of which air currents are blown. In boiler work, furnaces surrounded by tubes connected to the boiler circulation have been tried, but where the tubes are on the inner side of the walls, they have a tendency to absorb too much heat and cool the furnace temperature below the efficient point. With circulating tubes properly located in the brick work the heat absorbed from the furnace would be much less and it should be comparatively easy to obtain the desired temperature ratio between walls and furnaces. It is needless to say that consideration would have to be given to provision for cleaning and removal of these circulating tubes without disturbing the brick work.

The blowing of air against thin side walls and allowing the heat to dissipate in the room would greatly increase the radiation losses with consequent loss in efficiency. This difficulty, however, can easily be obviated by building the walls hollow and circulating the air for combustion through the hollow walls. This system has been used very successfully on both powdered-coal and stoker installations and has the very decided advantage of raising the furnace temperature and at the same time cooling the walls and reducing radiation losses.

Another difficulty frequently met with in wall construction is the lack of uniformity in dimensions of firebrick, thus necessitating

the use of thick joints; the mortar used being weaker than the brick offers a starting point for erosion. This difficulty is overcome in some instances by grinding the brick to a true surface and dimensions.

The effect of the life of furnace brick work on a boiler plant cannot be considered entirely from the point of view of efficiency because the cost of brick-work repairs and the time lost due to these repairs are large items in the cost of generating steam.

E. N. Trump. A survey made by the Fuel Administration during the war showed that the *average thermal* efficiency of the boilers in New York state could not be more than 50 per cent even though large plants were operating at 80 per cent efficiency. Conditions are not much improved today.

While we should study the investment factor in determining whether or not to replace an inefficient plant we must remember that the supply of coal in this country is not inexhaustible, that it will always increase in cost and that we owe something to posterity. We can not expect to go on taking more than we give, as we have done with lumber. We should be content with a low rate of return on the capital invested, expecting other advantages more difficult to calculate.

With a steady load, cast-iron economizers placed so as to have as short and direct flues as possible will add 7 per cent to the efficiency when the heating surface is 50 per cent of the heating surface of the boiler with a steam pressure 165 lb. gage.

With water at 120 deg. fahr. and out-going gas at 300 deg. fahr. even with a moderate amount of sulphur in the coal corrosion will take place rapidly and cast-iron tubes will not last more than eight years. In a waste-heat boiler with steel drum and tubes making steam at 10 lb. pressure and fed with water at 140 deg. fahr. it was found necessary to replace the tubes every three years and the drum in ten years. The fuel was Hocking Valley coal which is rather high in sulphur. The use of wet steam in soot blowers for economizers causes corrosion; it would be better to use air or superheated steam. Washing down the economizers with lime water or with a solution of soda to neutralize the acids as soon as they are taken off and while they are hot would probably reduce the action of the acids on the tubes.

When very large boilers are used the load factor of the whole plant is less than with smaller units. Trouble with economizers seldom comes at the same time as trouble with boilers. If the boilers

are well arranged it may be possible to use a by-pass with a little extra flue or loss. The best arrangement the writer has seen of type W, B & W boilers, is two economizers erected over each 2300-hp. boiler. The gas passing out of each side of the boiler enters the economizers from opposite directions and passes out to separate flues above the boiler on each side. A direct connection is made to the flue above the boiler flue, on the entering end of each economizer, with a set of dampers which cut out one economizer at a time.

As the load factor of a boiler house increases the economical rate of operation increases, but this also changes with the cost of fuel and the rate of wages as well as the assumed rate of depreciation and interest on the plant. For continuous operation with a load factor of 90 per cent the writer has calculated that for a first-class boiler house one can not afford to run above 150 per cent of rating without carefully considering an addition to the plant. If, however, the load factor is only 45 per cent as is often the case in a large power plant the portion of the cost of steam which is a charge for interest and depreciation on the plant will be doubled and this will considerably increase the economical rating. For example, if the cost of steam is 50 cents per thousand pounds with coal at six dollars per ton of 2000 lb. and the evaporation is ten pounds of water per pound of coal the cost of the coal will be 30 cents per thousand pounds of steam. The labor and repairs will be 15 cents and the interest and depreciation 5 cents with 90 per cent of the boilers operating at 150 per cent of rating. If the amount of steam is reduced to one-half the above amount the cost of steam will be increased five cents per ton by the interest and depreciation alone without taking into account the stand-by losses, in increased labor and repairs. By forcing the boilers during the peak load so that the total steam per twenty-four hours is the same as before, the losses are reduced to stand-by losses when running slowly and the decrease in efficiency at higher rating will be offset by less depreciation on smaller cost of plant. It becomes necessary to figure each case for the condition likely to occur and this becomes very complicated.

The first boilers of the B & W type which the writer had charge of were set with the tubes not more than three feet above the fuel bed. Of some ten installations in the same plant put in at about equal intervals in 40 years each has been set higher until 10 or 12 feet is now the rule. The type W with its huge firebox and great radiating surface has a decided advantage over any other type and

when properly operated will give with a steady load an efficiency of 80 per cent for boiler and stoker furnace and 7 per cent additional for economizer.

The form of furnace used in the type W boiler has such a large volume and provides such a good mixture that CO_2 can be carried higher without CO than in any other type with which the writer is familiar. The side walls must be built with a camber and of the maximum thickness which the brick will stand to prevent radiation losses. Air-cooled boxes on each side of the stoker 12 or 15 inches wide so that the coal from the retort nearest to the wall will overflow on it without touching the brick work eliminate the sticking of clinker to the side wall and the destruction by breaking it off. Slagging of the boiler tubes will occur either with stoker or powdered fuel if the furnace is too small but not with a properly designed furnace.

The writer is glad that Dr. Jacobus has emphasized the need for high-grade instruments and apparatus. Good automatic damper regulators, CO_2 recorders and thermostats with a competent crew to run them will pay a large interest on their cost, and supervision of the operation by an expert combustion engineer will give a surprising return.

Why is it that so many large concerns will spend great sums on an excellent engine plant maintained in the best style and carefully watched for small losses but entirely neglect the boiler plant where the losses may be ten times as great as the possible savings in the engine room. The Society is to be congratulated in having received such an able paper as Dr. Jacobus has written. It is filled with the wisdom of long experience.

ALBERT A. CARY. Dr. Jacobus' paper is not only timely, but instructive and of considerable interest to those installing and operating steam generating equipments and it is certainly worthy of careful perusal and consideration.

Referring to Par. 27 regarding the separation of boiler and furnace efficiencies; the writer is ready to agree that it is impossible to determine the efficiency of any one design of boiler so as to state its relative efficiency as compared to any other design or type of boiler.

The individual or separate efficiency of the boiler, *per se*, depends wholly upon the total amount of heat it receives from the furnace as compared with the total amount of heat it rejects at its gas outlet.

Generally speaking, if the same weight of gaseous products of

combustion is delivered from the furnace at a higher temperature in one case than it is in another case, a higher individual boiler efficiency may be looked for with the hotter furnace gases, as it is generally conceded that the rate of heat transmission varies in a greater proportion than the direct difference in temperature between the furnace gases and the water in the boiler, so that the rise in temperature of the escaping gases is by no means in direct proportion to the increase in furnace temperature.

This statement points to the fact that, under ordinarily good conditions, the higher the temperature obtained in the furnace, the higher the individual boiler efficiency becomes.

The proper design of the boiler including the arrangement and area of its heating surface to properly receive and absorb the heat delivered to it, the design and conditions of its surrounding setting as well as its arrangement and condition of baffles, the cleanliness of its exposed heating surfaces, both inside and outside, affect the separate efficiency of the boiler (considered apart from the furnace), thus making it practically impossible to state definitely the individual efficiency of the boiler under all conditions.

It is, nevertheless, possible to determine the efficiency of the boiler *per se*, with a satisfactory degree of accuracy under the conditions existing during a specific test, and the writer has frequently found it very useful to do this, so as not to charge up the sins of a poor and inefficient furnace against the operations of an excellent and efficient boiler, or vice versa, which is not found by merely taking the combined efficiency, and desirable knowledge is obtained as to whether the boiler itself is operating under the best conditions.

There is certainly a distinct difference in the functions of the boiler and furnace.

As a matter of fact, the boiler is essentially a piece of physical apparatus, built to accomplish the physical action of heat transfer; whereas the furnace may be regarded as a piece of chemical apparatus, built to accomplish the chemical reaction between the atmospheric oxygen and the combustible constituents of the fuel fed to the furnace.

As the individual efficiency of the boiler depends to a large degree upon the individual efficiency of the furnace, it is in many cases, important to determine what the individual efficiency of the furnace really is under its different rates of operation.

During the course of his experience, the writer has been called upon to test furnaces used for other purposes than the generation of

steam, and the presence of a boiler, with which to measure the heat production, has not been found necessary to determine the efficiency of these furnaces.

But, on the other hand, with boiler furnaces, the expanded furnace chamber, extended to receive the boiler within its setting, the proximity of the comparatively cool boiler heating surface to chill the burning furnace gases and suppress combustion, which suppression is frequently followed by secondary combustion referred to by the author, and a poorly designed combustion chamber (as regards both volumetric space and length of gas run), all contribute to the inefficient operation of the furnace, which facts make the boiler-furnace efficiency determinations individual and pertaining alone to the specific conditions found in the particular setting being tested.

Notwithstanding these limitations the writer has found it of great value to determine the individual boiler-furnace efficiency, which information has indicated where faults exist and where and how improvement may be obtained.

In determining furnace efficiency, it is essential to collect samples of gas for analysis as near the position where they leave the furnace proper as possible.

The term "flue gas analysis" is commonly used for samples collected generally. The writer makes a distinction between the two terms, "furnace gas analysis," which gives the conditions of combustion in the furnace and its immediately connected combustion chamber, and the "flue gas analysis," which is principally useful to show the amount of air infiltration through the setting as well as the effect of secondary combustion between the furnace and the flue.

In most of his work, the writer collects simultaneous samples from both the furnace and a position within the setting, just back of the flue outlet.

These samples are withdrawn in a constant continuous stream by means of the suction apparatus, and they are passed through a special gas collector, from which fresh gas samples of exactly the same composition as exists within the setting at the moment they are required for analysis are made easily available.

The remarks made by the author concerning furnace brick work are very important.

With the considerable height to which boilers are being built at present the required increase in heating surface is obtained, but

without a corresponding increase in the furnace area, and as our boiler heating surface increases, a greater amount of fuel must necessarily be burned per square foot of grate surface per hour.

With this increased rate of combustion, high temperatures are obtained and the difficulty in maintaining firebrick linings in furnaces is greatly increased.

The selection of refractory material for lining furnaces is a matter which has received too little attention by power-plant owners, who are very apt to leave the matter of its selection to their masons who are, in many cases, pretty certain to buy the cheapest firebrick they can purchase or else buy at places where their commissions are the largest.

The term "firebrick" means little to those who have a knowledge of refractories, as it includes a considerable variety of material and covers a "multitude of sins."

A common practice of using the same quality of firebrick for all parts of the furnace is certainly a mistake.

The side-wall linings of a furnace should be selected with the use to which they are to be subjected kept in mind.

It is true that these side walls are subjected to the effect of high temperatures, but if we use the most refractory brick we can buy for this position we shall find that they will not stand the severe abrasion to which they are subjected, such as received with the use of a poker, a chisel, the clinker bar, etc., to free them from the adhering clinker, and it is most important also to consider the fluxing effect on these bricks from the ash produced from certain kinds of coal.

As stated by the author, a fire-clay brick is the one used in nearly all boiler settings, and it is probably the brick best adapted to sudden changes of temperature, due to its more or less porous structure.

It is composed of a mixture of two varieties of clay, the hard flint clay forming its highly refractory body, while a soft plastic clay is mixed with it to act as a bond and give it strength.

As these clays are obtained from various parts of the country and in various degrees of purity, and as the manufacturers mix them in different proportions, we have a considerable variety in grades of firebrick to select from.

The finely ground, hard-burned brick usually makes the best lining for side walls; while the coarser or more open (or porous) bricks are better adapted for fire arches.

The difference between plastic deformation and the melting point of firebrick is well brought out by the author, as many are misled by merely taking the melting point alone in considering the selection of firebrick, and the temperatures given by him correspond very closely to the temperatures the writer has obtained in tests.

The one point previously referred to in connection with firebrick linings which is often overlooked is the effect of some ash having certain compositions producing a fluxing effect upon the firebrick lining when they come together at high temperatures. Some firebricks and ashes, when raised to certain temperatures, have a fluxing effect one upon the other, and the writer has found certain firebricks having a comparatively high melting point which readily fused in contact with ash at a much lower temperature, viz. 1700 deg. fahr., causing a considerable erosion in the linings.

F. F. Uehling. The author states in Par. 24 that a small amount of CO is often present when a high percentage of CO_2 is carried. He also gives a definite example of what a certain percentage of CO means in increased loss, as compared with the increase in the percentage of CO_2 necessary to offset the CO loss.

Since the relation which the percentage of CO in the products of combustion bears to the loss up the chimney is a very interesting one, it occurred to the writer that a few curves showing chimney losses for different percentages of CO and CO_2 might prove not only interesting but also of value to engineers when the importance of CO content in flue gases is given their consideration.

One of the most striking facts brought out in the curves is that there is not only a *loss* due to the presence of CO but *also a gain*, and the *net magnitude* of the CO loss is their difference. In other words, the effect of CO on chimney loss is twofold, one positive and the other negative, and the resultant actual CO loss can be determined by their algebraic summation. Furthermore, the net effect of CO on the chimney loss may be *plus*, *minus* or *zero*, depending upon the percentage of CO_2 and the temperature of the escaping gases.

What actually takes place when CO is present is:

First, a loss results due to the unliberated heat in the CO itself, that is to say the additional heat that would be available if the CO in the products of combustion had been oxidized

Second, a gain occurs due to the fact that the more CO there is in the products of combustion for any given percentage of CO_2 the less will be the weight of the products of combustion per pound of carbon burned and consequently the less will be the sensible heat dissipated by the flue gases as they leave the boiler.

The solid lines of Fig. 3 represent the *total* sensible and unliberated heat losses up the chimney for different percentages of CO

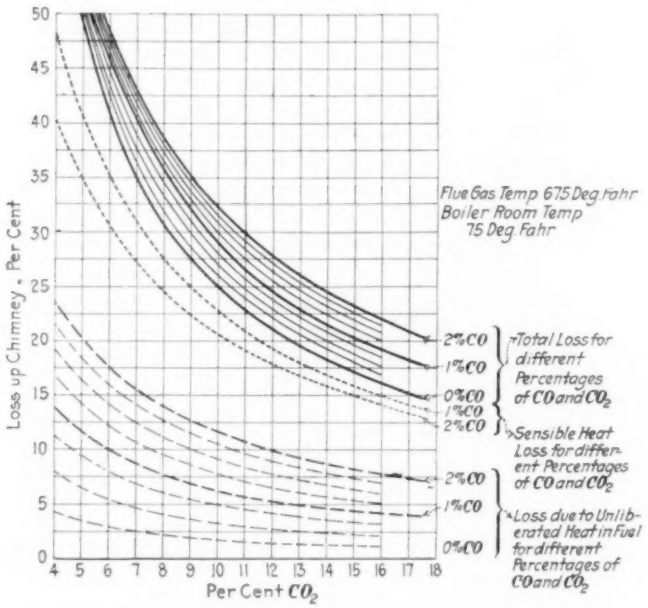


FIG. 3 TOTAL SENSIBLE AND UNLIBERATED HEAT LOSSES, FLUE TEMPERATURE 675 DEG. FAHR.

and CO_2 when the flue gases from carbon fuel escape at a temperature 600 degrees above atmospheric temperature. Fig. 4 gives corresponding information when the flue-gas temperature is 400 degrees above atmospheric temperature.

It is very easy to determine from these curves how far it pays to go in increasing CO_2 in the presence of CO. For example, referring to Fig. 3, 9.5 per cent CO_2 with no CO represents a loss of 27.5 per cent, whereas 1 per cent of CO with 10.5 per cent of CO_2, also 2 per cent of CO with 12.25 per cent of CO_2, represents the same loss per pound of fuel burned.

Fig. 5 is based on flue-gas temperature 600 degrees above atmospheric. It shows how much the CO_2 would have to be increased in order to counteract the increased loss caused by the presence of CO.

Figs. 3 and 4 illustrate two other very important facts, viz.:

a The *loss* due to the unliberated heat in the CO contained

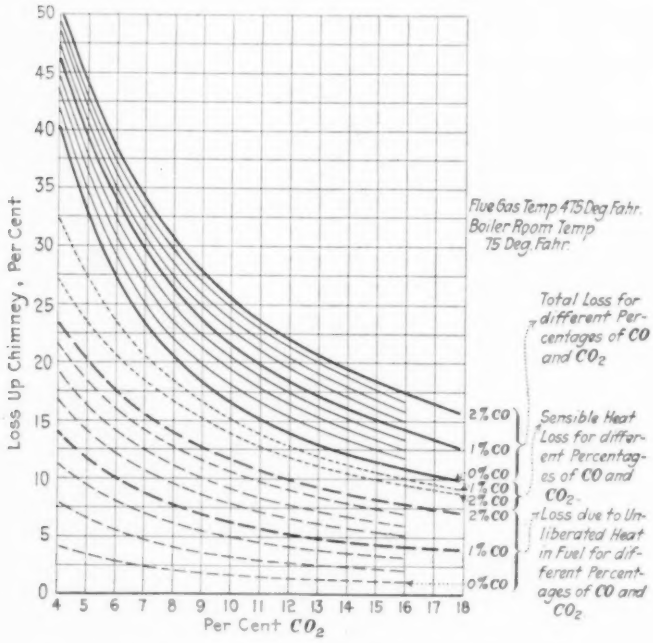


FIG. 4 TOTAL SENSIBLE AND UNLIBERATED HEAT LOSSES, FLUE TEMPERATURE 475 DEG. FAHR.

in the products of combustion is constant for any given percentage of CO_2 and of CO regardless of the flue gas temperature

b The *gains* due to CO are greatly increased with increase of flue-gas temperature, for any given percentage of CO_2 and of CO.

It follows, therefore, that temperature is a factor which *must* be taken into consideration when the actual CO losses are to be determined.

WM. S. ALDRICH. Our observations refer to the performance of a common type of vertical water-tube boiler under rather abnormal conditions, which seem to confirm the conclusions drawn by the author from normal practice, and the principles established thereon for improvements.

The heat absorption of the boiler is almost entirely from con-

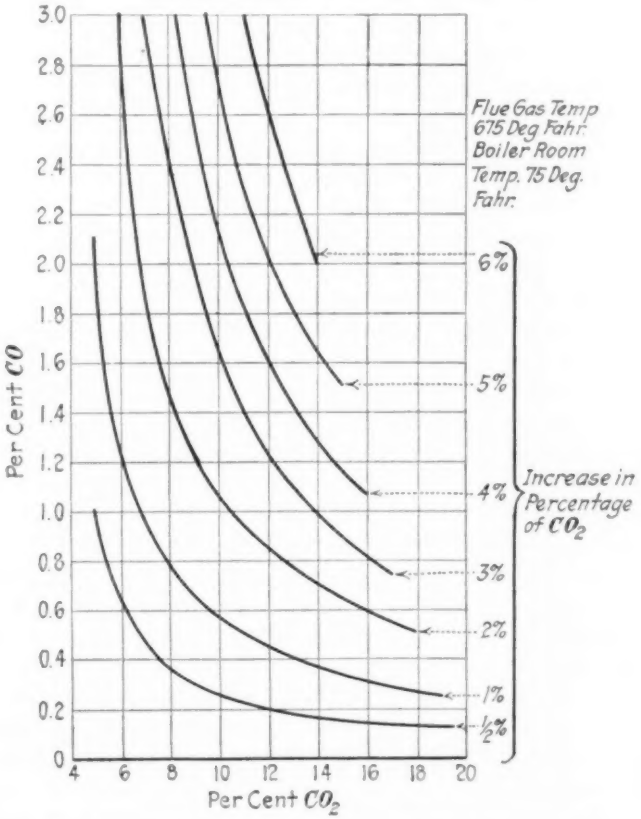


FIG. 5 INCREASE IN CO_2 NECESSARY TO COUNTERACT THE INCREASED LOSS DUE TO CO

vection, owing to the nature of its original installation, as a waste-heat boiler for a forge-heating furnace. As a hot-gas type of boiler, or convection boiler, it has developed a maximum 92 per cent of rated capacity, utilized about 70 per cent of the heat delivered by the escaping gases on their way to the stack, and accounted for about

95 per cent of its evaporation through heat absorption from convection, allowing the remaining 5 per cent absorption from radiation.

Conditions arose requiring the use of fuel oil, and a combustion chamber was sealed off from the main furnace of about 385 cu. ft. capacity. Whether a coincidence or otherwise, the above performance was practically duplicated with fuel-oil burning at a rate of from 90 to 100 gal. per hr. The fuel oil used was 35 deg. B., weight about 7.1 lb. per gal., heating value of 19,680 B.t.u. per lb., and practically a pure hydrocarbon. This was an unusually large combustion space, according to the old usage of about 1.85 lb. oil per cu. ft. against the expected 5 lb. allowed. The whole gamut was run of all the experiences cited by the author, and a few others, in a twenty-foot throw or trajectory of the oil flame, — first a flat flame and later a cone flame, — before its exit from the Dutch oven, at the base of boiler, as a hot gas suitable for steam generation by convection.

It was necessary, therefore, to obtain a maximum difference of temperature between the entering and exit gases of the vertical boiler. This was reached at about 2250 and 400 deg. fahr., respectively, but it could not be maintained for the evident reasons indicated by the author, namely, "some efficiency must be sacrificed in most cases in order to maintain the furnace brickwork."

The larger difficulties seem to come from erosion; and it is even yet an open question if the products of combustion in fuel-oil burning do not produce the greatest trouble with certain refractories which otherwise may be entirely satisfactory in meeting temperature requirements and resisting spalling action. Protecting the joints by a highly refractory wash is more or less in vogue. This seals the furnace quite effectually against the leaching outward of the hot gases, while an effort is made to increase its performance in the only way possible, when absorbing the heat by convection; namely, to check the draft and to reduce the excess air to a minimum.

Another series of trials was made with this unit as a direct-heat coal-fired boiler, by sealing up the heating furnace attached, and opening a five-foot semi-circular fire archway between this and the gasway leading to the Dutch oven at base of the boiler. This gave about 1200 cu. ft. of furnace volume; and as high as 375 boiler horsepower has already been developed. This is 150 per cent of the steaming capacity with an allowance of 4.8 cu. ft. of furnace volume per rated horsepower. The important feature to be emphasized is that this is still a convection boiler, for no change that has been made

offered any possibility of increasing the absorption from radiation. The coal used was from Franklin Co. (Ill.) field.

DAVID MOFFAT MYERS. The question which arises in the mind of the writer is how to apply broadly the useful knowledge which engineers possess on the subject of the conservation of fuel, and the answer he has arrived at is that conservation of fuel cannot with safety be left in the hands of the individual. In the past the individual engineer has wasted our oil supply to the extent that 50 per cent of the oil taken out of the ground has been wasted. It is the same with coal. The reason for such waste is that the individual having charge of the oil and the coal will use it in a way to bring him the greatest financial return. In the same way the owner of a plant considers that he can make enough money without bothering about conservation in a boiler plant.

This matter cannot be left to the individual; it is of community interest. The public at large does not fully realize the importance of fuel as a national issue although it is coming slowly into such a realization. Therefore although the time is not yet ripe for any radical steps we must look forward ultimately to a solution of the fuel problem from the standpoint of community interest. In former days, before the enactment of compulsory boiler safety laws, owners of boilers considered government supervision and inspection as impertinent. Such supervision is now taken as a matter of course. The public must be made to see how adversely it is affected by the great and uncontrolled loss of fuel, so that government coöperation and assistance will be demanded.

As a measure looking toward such service are the possibilities of fuel conservation by the Bureau of Mines which involve the necessity of educating the public to the extent of the wastes. Great advantages are to be secured by the appointment of a resident engineer in each state to keep in touch with local industries and with a clearing house of information in Washington. A body of citizens advising the Bureau of Mines and representing business men, engineers and citizens from various parts of the country should assist in planning and developing the conservation program.

There is necessity for supervision in the correct design of new power plants and also in changes in power equipment in existing plants. Better educational measures should be adopted providing courses on fuel conservation in technical schools and for engineers and firemen. There should also be examinations for licenses in this

profession and certificates provided for engineers and firemen so educated.

C. A. Frankenhoff. Is the average temperature of the firebrick mentioned in the paper the temperature of the surface of the walls or the average temperature of all the firebrick itself?

Have any definite tests been made to determine the proper design of furnace wall as to best weight or thickness and chemical composition of the brick, taking into consideration the best boiler efficiency and the lowest maintenance cost?

John A. Stevens. In the new central power station in Paris which is supposed to be of the most modern design the air is preheated to 190 deg. fahr. This is done beyond the economizers and the temperature of the gases rejected to the stack is about 200 deg. fahr. Another system of preheating is by using steam at 170 deg. It would seem, therefore, that the proper temperatures lie somewhere between 170 and 190 deg. The preheaters consist of a series of plates located between the economizer and the stack so arranged that the exhaust gases pass through one division and the air through another. They are easily cleaned both inside and outside.

G. L. Warden. The writer has found that while industrial power plants might be operated at high efficiency the losses in the various operating departments of the industry may be very considerable. By the installation of a metering equipment so as to know exactly what the power losses are it has been possible to reduce them by 25 per cent, thus justifying the installation of the meters, as well as reducing cost per unit.

R. Sanford Riley. Dr. Jacobus states: "for securing the highest efficiency, the furnace temperature should be the maximum that can be obtained, and combustion should be completed within the furnace chamber."

We all agree that combustion should be completed within the furnace chamber. This requires volume and proper opportunity for mixing of the gases. The writer cannot see, however, that efficiency is increased by the maintenance of a high temperature in the furnace between the fuel bed and the boiler. The maximum temperature due to combustion of the fuel will be obtained when there is a high percentage of CO_2 without CO, or, in other words, when there is a minimum of excess air accompanied by complete combustion. The above condition represents, of course, the highest

combustion efficiency and if Dr. Jacobus means "temperature of combustion" or "fuel-bed temperature" when he says "furnace temperature," the writer can certainly agree with his statement. The temperature within the furnace chamber, however, need not be as high as this maximum temperature of combustion.

Arches or walls so constructed that they deflect or hold the heat, tend themselves to become as hot as this maximum temperature of combustion. If these special arches or walls are not needed as an aid to combustion, they are, in the writer's opinion, detrimental and he should like to know if Dr. Jacobus claims any advantages for them. Dr. Jacobus has very properly pointed out their disadvantages in connection with maintenance of brickwork and also in slagging of coal which has ash of low fusing temperature.

Elsewhere in the paper Dr. Jacobus mentions the large utilization of radiant heat, and the writer is much impressed with the statement that B & W boilers may absorb as much as 60 per cent of the total heat by radiation. It seems to the writer that this large absorption by radiation is all to the good, and the consequent lowering of furnace temperature is one reason why it is so good. The writer sees no compromise necessary in this respect, except to secure ignition. With underfeed stokers there is no need to make any compromise whatever, and the best practice seems to be to expose the maximum area of tubes to radiant heat and thus hold down the furnace temperature. This transfer of heat by radiation is the direct route "from maker to user." There is then no middleman like brickwork to take toll and cause trouble. Convection can also be considered a middleman but does not require much attention or cause any trouble.

The writer suggests the theory that all boiler-furnace construction should utilize radiation to the maximum, and so hold furnace temperature down to the minimum. This theory has nothing to do with the temperature of combustion in the fuel bed, which, of course, should be the highest possible to correspond with the minimum of excess air.

THE AUTHOR. I wish to thank those who have taken part in the discussion. The information submitted adds greatly to the value of the paper. Dr. Goss's remarks are deeply appreciated as coming from one who is a past-master in the art and whom we all respect for the knowledge he has contributed to the subject.

Mr. Bailey states that trouble may result through the use of

air heaters through the collection of a pasty acid-bearing soot from certain fuels which it would be almost impossible to remove by ordinary means, and which would corrode the surfaces very rapidly at the comparatively low temperatures of the gases. The problem of resisting corrosion in an air heater is more difficult than in an economizer as the lowest temperature to which the surfaces are brought cannot be governed as readily as in an economizer. In economizer practice the temperature of the entering water may be made, say, 140 to 160 deg. fahr. as a minimum and the greater part of the external corrosion can be overcome thereby through preventing the condensation of moisture from the gases, or what is ordinarily termed a "sweating action." In the case of an air heater the air may enter under winter conditions at or below the freezing point even if taken from above the boilers and the temperature of the surfaces of the air heaters which come in contact with the flue gases leaving the heater would be somewhere between the freezing point and the temperature of the flue gases, or well below the sweating point for certain fuels and conditions of firing. The method of best handling this feature has not been worked out. It would seem advantageous, however, to arrange an air heater so that the gases leaving the heater flow downwardly, which would cause any moisture condensed from the flue gases to drip away from the heater instead of allowing it to run back and collect in the heater.

Mr. Bailey is correct in saying that probably the greatest need for missionary work is in connection with managers and chief engineers who do not appreciate the opportunities for savings in power-plant operation through the personnel in charge of the boiler plants. He is certainly right in his idea that automatic systems for combustion control will never take the place of brains, and while they function fairly successfully if left alone, the best results can be secured only by proper supervision and care.

Mr. Hopping brings out the advantages of large furnace volumes, and points out that aside from the additional efficiency secured through complete combustion in the furnace there is an added advantage at his plant through eliminating the cinder nuisance. He also speaks of the trouble experienced in some instances through slag adhering to the exterior of the boiler tubes, this slag coming from small particles of fuel and ash driven off from the fuel bed and carried with the gases up to the tubes, and of the advantages of a large furnace volume and a sufficient distance from the fuel bed to the tubes in minimizing the trouble. Trouble through slag depositing

on the outside of the tubes is greatly influenced by the class of fuel burned and especially by the rate of combustion and the method of burning the fuel. Arranging the lowermost tubes to give a free flow of the gases between the tubes to form what is called a slag screen has been found to give good results for certain conditions of service.

Mr. Hopping asks what temperature is safe for preheating air for combustion with some of the principal grades of fuel when the furnace is constructed of the best grade of firebrick. It is impossible to answer this question by setting a definite temperature as it depends on the kind of fuel, the method of burning the fuel, and the way in which the furnace is constructed. The usual method of failure of brickwork through a high temperature comes from the walls bulging inward toward the furnace. A good form of construction for resisting high temperatures is one where the furnace walls slope outwardly so as to form a hopper-shaped furnace. If the walls are sufficiently inclined the brickwork may soften considerably without failing. Again, the furnace temperature depends on the amount of boiler heating surface exposed to the radiant heat. With a furnace, stoker and boiler setting similar to those at Mr. Hopping's plant, with the furnace built of the best grade of firebrick, I believe that no prohibitive amount of trouble would result through preheating the air to a temperature of 300 to 400 deg. fahr. but experience alone can determine the limit to which preheating can be carried.

Mr. Ricketts brings out the advantages of a furnace which is narrower at the bottom than at the top. He also describes other methods such as air cooling for keeping the temperature of the brickwork below that at which it will deform or be attacked by the fused coal ash. He is certainly correct in saying that the effect of the life of the furnace brickwork on a boiler plant cannot be considered entirely apart from the efficiency, as the cost of brickwork repairs and the time lost due to these repairs are large items in the cost of generating steam.

Mr. Trump's data respecting the life of cast-iron economizer tubes for coals having a sulphur component, and his suggestion that the economizers be washed down with lime water or a solution of soda to neutralize the acids as soon as they are taken off, are useful and interesting. Where economizers are washed down with plain water it is highly essential that they be washed thoroughly, as, should all of the acid-bearing soot not be removed, the wet soot which will remain will have a highly corrosive action. It is further

advantageous to fill economizers with hot water to thoroughly dry the exterior as soon as possible after the period of washing. It is also well to place economizers in service within a reasonable time after they are washed to avoid the collection of any moisture on the exterior. It might be a good plan to first wash the economizers thoroughly with plain water and then with a solution of soda as suggested by Mr. Trump.

Mr. Trump's conclusion that a first-class boiler house cannot afford to be run at over 150 per cent of rating with a load factor of 90 per cent without carefully considering an addition to the plant, is a valuable one. This in connection with his statement that while we should study the investment factor in determining whether or not to replace an inefficient plant, we should remember that the supply of coal is not inexhaustable and we owe something to posterity is a sound argument for the desirability of carefully considering loss factors before deciding to run a plant at high capacities. Of course, the problem, as pointed out by Mr. Trump, is a different one when it comes to a low load factor and when there are peak loads of comparatively short duration to be carried. The operating efficiency obtained by Mr. Trump of 80 per cent for the boilers and 87 per cent for boilers and economizers shows what can be accomplished through a careful study of plant design and operation.

Mr. Cary considers it possible to determine the efficiency of the boiler *per se* with a satisfactory degree of accuracy under the conditions existing during a specific test to determine whether the boiler is operating under the best conditions. To do this he suggests making flue gas analyses of the samples of gases taken from the position where they leave the furnace. In most cases it is impossible to obtain a correct average analysis of the gases leaving the furnace on account of the variation in the samples taken at different points. In some instances unconsumed combustible gases enter the spaces between the tubes at one point and considerable excess air at other points, and the samples will be found to vary widely. An average at the different points will not give a true average analysis, as the velocity of the flow of gases varies at the different points. The best use that can be made of analyses of the sort is to endeavor to so fire the fuel that the results will be more or less uniform and in this way improve the combustion and the efficiency.

The effect of CO in the flue gas analysis as brought out by Mr. Uehling is certainly most interesting and useful. The problem is complicated through an additional element to those considered

by Mr. Uehling, namely, that a change in the flue-gas analyses results, with other conditions remaining the same, in a change in the flue-gas temperature.

Mr. Myers states that the conservation of fuel can not with safety be left in the hands of the individual as it is of community interest. His patriotic work in connection with fuel saving during the war will always be remembered. His efforts resulted as we all know in a vast saving at a time when everyone appreciated its necessity. The same necessity still exists, not as a war-time measure, but, as he states, as a community interest, and we should all endorse his suggestion that more study be given the subject.

Mr. Frankenhoff asks whether the average temperature of the firebrick mentioned in the paper is the temperature of the surface of the wall, or the average temperature of the brick itself. The temperatures mentioned in the paper were determined by allowing the brickwork to heat up slowly in a soaking pit, the exterior temperatures being determined as well as the interior temperature of the column, and the temperatures therefore represent the average temperature of the brick itself. Mr. Frankenhoff's second question as to whether any tests have been made to determine the proper design of furnace wall as to weight and composition of the brick, taking into consideration the boiler and the lowest maintenance of cost, broadly covers the field of boiler-furnace design and construction. Many are working on this problem which will not be fully answered as long as there is progress and improvement.

Mr. Stevens describes air heaters made up of plates and brings out the fact that they should be arranged so that they can be readily cleaned. It is questionable whether a plate heater or a tubular heater is preferable for most constructions, but with either construction there should be provisions for thorough cleaning.

Mr. Warden brings out the fact that there are often losses in industrial power plants entirely aside from the boiler room and that it may be possible to effect savings as high as 25 per cent through checking up and conserving the use of steam. This is an important element which is often lost sight of. In some cases most careful efforts could not result in raising the efficiency of the boiler room by more than a few per cent, whereas should the steam be used in the proper way it might result in, say, five or ten times any saving possible in the boiler room.

Mr. Riley suggests the theory that all boiler-furnace con-

struction should utilize radiation to the maximum provided this does not interfere with the combustion. As pointed out in the paper if an effort is made to absorb too great an amount of radiant heat directly from the fire, the completeness of combustion may be interfered with and the reduction in furnace temperature may offset some or all of the gain due to the direct absorption of the radiant heat. For example, it would not be possible in certain cases to burn wet sawdust fuel without employing reflected heat from the brickwork to bring up the temperature of the furnace and fuel bed. Starting from this particular case, where it is plainly evident that too great an absorption of heat by radiation would be a detriment, there are intermediate conditions where a balance must be secured between the amount of radiant heat absorbed directly by the boiler and a furnace temperature which will give the best combustion results.

Even where the combustion of fuel is within a more or less enclosed chamber, a great proportion of the heat is absorbed in the form of radiant heat, as the walls of the chamber and the walls of the setting where the hot gases are conducted to the boiler radiate heat to the surface of the boiler.

There are conditions where better results may be secured through mounting a boiler directly above a furnace so as to absorb the radiant heat from the fire rather than by mingling all of the gases within the furnace and passing them through a connecting flue to the boiler, these conditions being where the fire in one part of the fuel bed is at a much higher temperature than the other. In such a case if a boiler arrangement is provided where the radiant heat from the hotter part of the fuel bed is directly absorbed by the boiler it might give a greater efficiency than would be obtained through an arrangement where the gases are mingled within the furnace and afterwards passed to the boiler. This results from the fact that the absorption of radiant heat varies as the fourth power of absolute temperature, and it is possible in a case of the sort to obtain better efficiencies by absorbing the radiant heat from the hot part of the fire before mingling the gases and reducing the general temperature of the gases.

Again, as pointed out in the paper, there is a limitation in some cases to the temperature to which a furnace may be raised with economy on account of the higher temperature leading to the fusing of the ash and clinker in the fuel bed. A high furnace temperature may also result in a greater amount of difficulty through the accumu-

lation of slag on the lowermost tubes than would be experienced with a lower furnace temperature. The fact remains, however, that in general the higher the furnace temperature the higher will be the efficiency. The problem is a complex one and can be dealt with only in a general way.

No. 1814

FUEL SAVING IN MODERN GAS PRODUCERS AND INDUSTRIAL FURNACES

By W. B. Chapman, New York, N. Y.

Member of the Society

This paper calls attention to the wastes of fuel in the industries using gas producers and producer-gas furnaces. Both of these problems are in the field of combustion engineering.

The progress made in the last twenty-five years in gas-producer construction is emphasized by the descriptions of leading mechanical producers now in the market, together with an indication of their fuel-saving possibilities.

Attention is called to the savings made possible by using suitable accessories in the gas house.

To illustrate the possible savings in furnace operation a description is given of a distinctive type of recuperative furnace. The extension of the use of such a furnace to pulverized coal and oil would, the author states, result in similar savings.

THERE are about 10,000 gas producers in the United States. With the exception of the modern mechanical producer, most of these are "sick" and badly in need of a "gas doctor," and the furnaces they supply are in an equally bad way.

2 The producers in use are divided approximately as follows: 6500 in the steel and allied industries; 1500 in the glass industries; 500 in the chemical industries; and 1500 in miscellaneous industries. Under "miscellaneous" are included the ceramic industries, lime burning and about 200 gas producers used for power. The gas producer for power purposes has never been much of a success — except for small units using anthracite coal.

3 Engineers have given much attention to the engine room, some to the boiler room, a little to industrial furnaces, and least

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of all to the making of raw producer gas. Hence the backward condition of the gas house.

4 No definite data have been compiled by the Government on the amount of coal used in gas producers, but with the assistance of

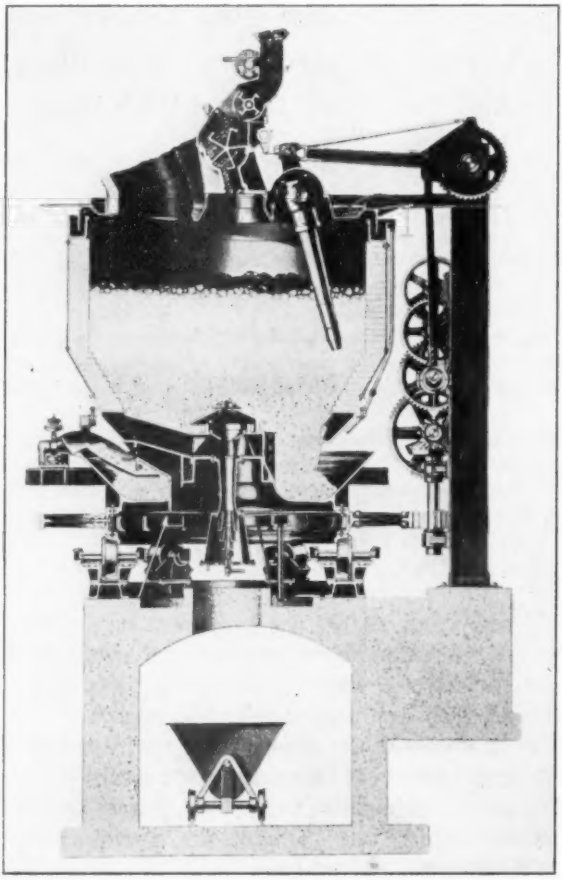


FIG. 1 HUGHES PRODUCER

such figures as are available it is estimated roughly that in the steel industries about 14,000,000 tons of bituminous coal are transformed annually into raw producer gas for use, and in the glass industries about 2,500,000 tons, and about the same amount in the various other industries.

5 In the steel industries approximately the same amount of coal is used for making gas to heat furnaces as for making steam. In glass making three-fourths of all the fuel is used in gas producers. But wherever producer gas is used it is apt to be the most backward part of the business. A given amount of time and money, if spent on improving conditions in the gas house, will usually bring

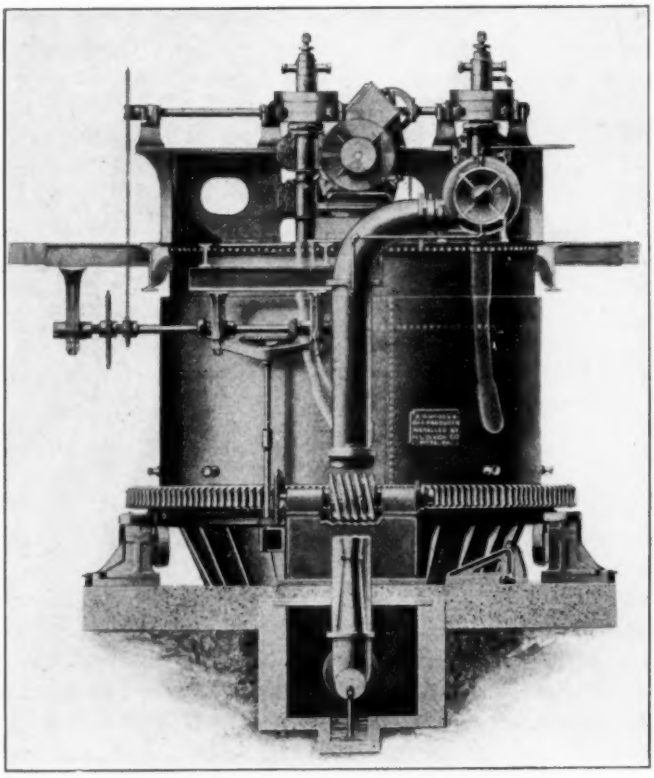


FIG. 2 R. D. WOOD PRODUCER

larger returns than in any other department. In most industries requiring large heating operations more trouble arises in that department than in any other part of the business.

6 Much progress has been made in the past 25 years in gas-producer construction. It started with the excellent work of W. B. Hughes. Progress in furnace construction, however, has lagged. The chief incentive to improvement of both producer and furnace

has been high-priced labor, and the few advances made in furnace construction have been confined mostly to certain labor-saving features. Now the situation has changed; fuel saving has been put on a par with labor saving, and interest is being noted all along the line.

THE PRODUCER-GAS PROCESS

7 In making producer gas there are three steps or operations: (1) Feeding the fuel; (2) agitating the fire; and (3) removing the ashes. Progress in producer construction has centered on various ways of performing these three operations automatically.

8 *Automatic Coal Feeding*, if continuous and uniform, will increase the B.t.u. in the gas about 10 per cent and will improve its uniformity to such an extent that it can be burned in the furnace with a 10 per cent improvement in economy. However, the labor saving due to automatic feeding is slight, and should not be confused with the labor saved by the use of coal-handling equipment and overhead bins, always a good investment for installations of more than one producer.

9 *Automatic Agitation*, if suited to the kind of coal used, will increase the B.t.u. from 10 to 20 per cent (depending largely upon the zeal and skill with which it was formerly hand-poked) and the uniformity thus obtained will cause a corresponding saving in utilizing the gas in the furnace. The combined effect of automatic feeding and automatic agitation usually makes possible a saving of 25 per cent of the fuel required for a given operation.

10 *Automatic Ash Removal* is of two kinds — intermittent and continuous discharge. The intermittent type removes the ashes once or twice in 24 hours. It is open to the objection that during the ash-removal period the fire is badly disorganized and a poor quality of gas is made for possibly half an hour. The continuous type of ash removal is entirely free from this objection. In Europe the continuous type is much in favor, although but few have been installed in this country.

11 The saving caused by mechanical ash removal depends largely upon the kind of device used. If the ashes are removed intermittently, and if no effective agitation accompanies the ash removal, the saving will be limited to the two or three hours' time required daily for the ash men to shovel the ashes out by hand. If, on the other hand, the ash removal is continuous and is accompanied by suitable agitation of the entire ash bed and lower portion of the

fire bed, there will be a considerable saving in the labor required both for manipulating the fire and for cleaning clinkers off the walls. Moreover, the continuous type of ash removal, accompanied by ash agitation, will increase the capacity of the producer about

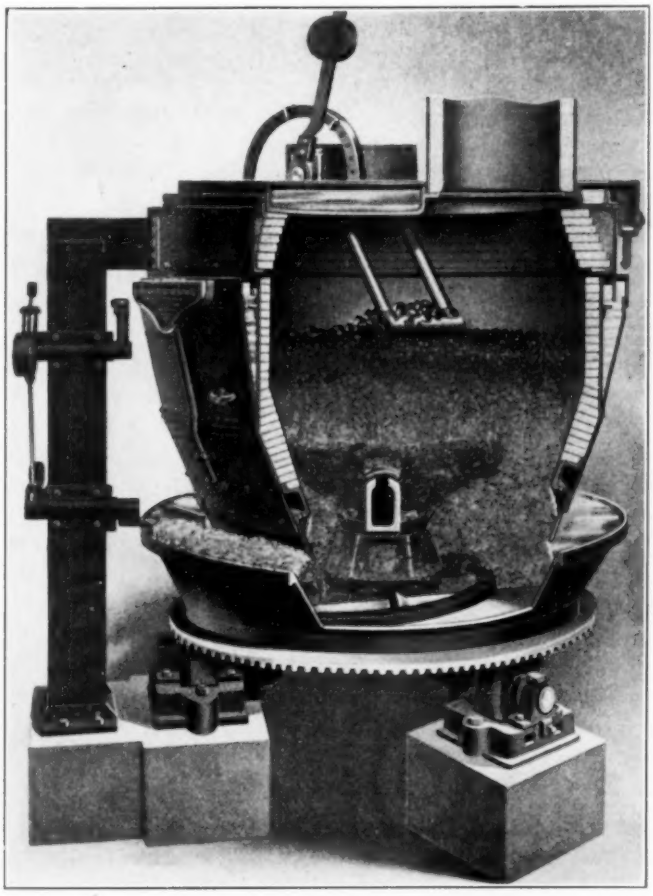


FIG. 3 MORGAN PRODUCER

50 per cent and will save some fuel on account of the improvement in the quality and unformity of the gas. Unfortunately the continuous type of ash remover, combined with effective agitation, is little known in this country, although it is the common type in use throughout Europe.

TYPES OF PRODUCERS

12 Bearing in mind the foregoing three steps in making producer gas, let us now consider the types of producers in most common use in America, the Hughes, the R. D. Wood,[1] the Morgan and the Chapman. Each is an excellent machine.

13 *Hughes Producers.* To Mr. Hughes, then chief engineer of the Pencoyd Iron Works, is accorded the credit for having installed at that plant in 1897 the first successful mechical producer in America. This machine is still in operation. As will be seen from Fig. 1, the chief feature consists in a vertical water-cooled finger hinged to the stationary top of the producer and made to oscillate between the center and the wall while the body of the producer and its contents revolve underneath. Thus, in time the entire contents of the producer are stirred. The speed of this producer was originally one revolution in 20 min., but every few years it has been increased until now the walls make one revolution in 8 min., with considerable increase in capacity.

14 Since the installation of the first Hughes producer, an automatic ash-removing device has been added. It consists in a stationary bar arranged to sweep the ashes from a revolving grate. The bar is inserted once or twice every 24 hr. for a short period and then removed. No attempt is made to use the device for agitating the lower portion of the fire bed. Recently an automatic feed has been added — in fact two automatic feeds — thus making the producer completely mechanical. The double feed gives unusually good coal distribution. This is the oldest producer on the market, and there are over 900 in operation — more than any other mechanical producer.

15 *The Wood producer,* Fig. 2, is somewhat like the Hughes, in that it uses the vertical-stirring-arm principle. There are two vertical arms, one near the center and the other near the wall. These stirring arms are bent as shown in the illustration, but instead of oscillating from center to side they revolve around their vertical axes. The walls of the producer revolve and thus the fire bed is carried past the stirring device. The coal is fed from a rotating drum located eccentrically. As the revolving fire bed comes under the feeding device fresh fuel is supplied to it. The speed of the producer is one revolution in 30 min. A steam-turbine

[1] The Wood type is manufactured by two different companies.

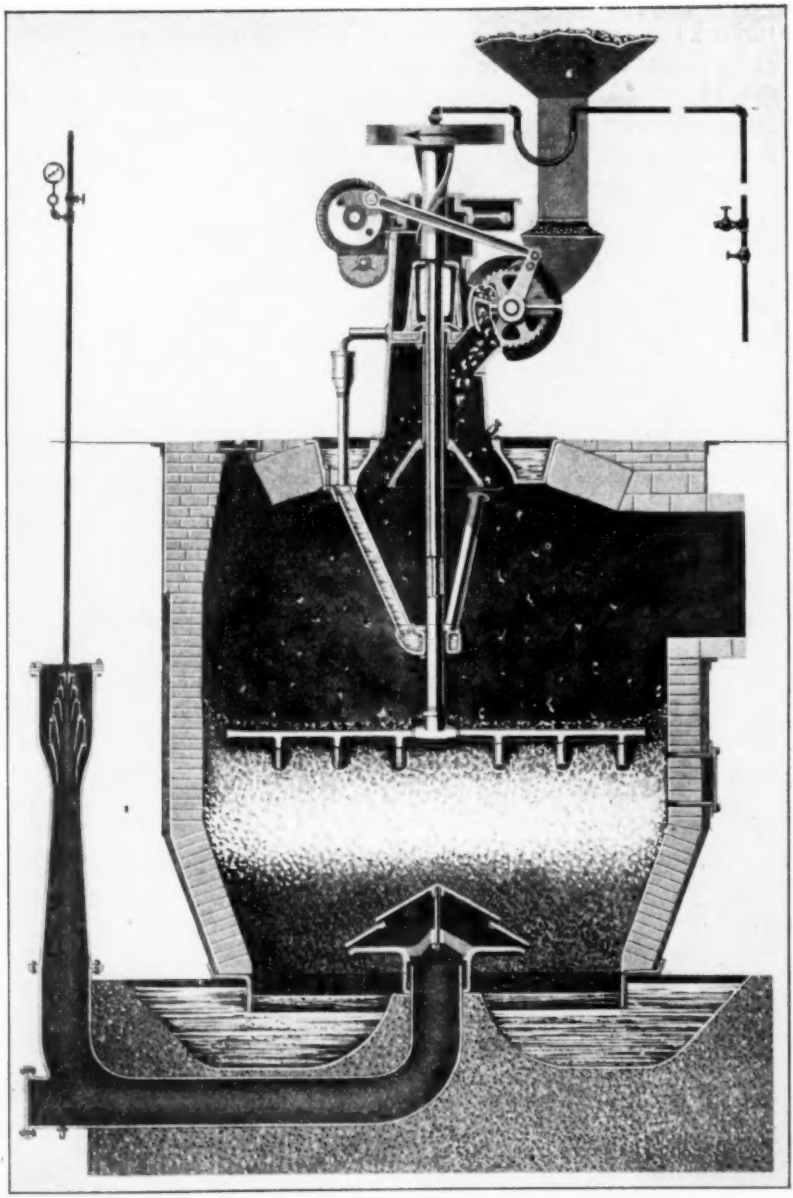


FIG. 4 CHAPMAN PRODUCER

blower is supplied in addition to the ordinary steam-jet blower. The turbine has an extra large capacity, operates quietly, and usually gives results that are more uniform than a steam-jet blower.

16 The ash is removed by a blade or plow attached to the lower edge of the revolving wall and extending down into the ash bed. The blade takes out ashes at a fixed rate, which is somewhat faster than they are made and therefore it cannot be operated continuously, although nearly so. Scoops are attached to the skirt of the revolving wall for carrying the ashes around to a suitable point for final discharge. The capacity of this producer is unusually large, owing to the facts that there are two stirring arms in place of one, and the removal of the ash is nearly continuous. Moreover, the diameter of the Wood producer is 10 ft. 6 in., instead of the usual 10 ft. This producer is particularly popular among glass manufacturers, and some remarkable records have been secured with it.

17 In a producer having vertical stirring fingers it is highly desirable to keep the top and bottom of the fire bed always at the same level so that the stirring fingers will always project the right distance into the fire bed for best results. If these fingers should reach too near the ashes, the air blast would break through into the recess or gap left in their wake and spoil the gas. A continuous, or at least semi-continuous, ash removal would thus seem to be necessary in order to keep approximately the same amount of ashes in the fire bed all the time and the same amount of ashes continuously in the producer. Producers with long vertical stirring fingers are difficult to operate with a thin fire bed or at less than two-thirds of full capacity.

18 *The Morgan producer*, Fig. 3, like the Hughes, is 10 ft. in inside diameter and has about the same capacity. Also, like the Hughes, it is very popular in the steel trade. However, it differs radically in both the method of agitation and in the ash removal. Instead of vertical arms projecting deep into the fire bed, a horizontal arm which rides on the surface is used, it being claimed that surface agitation is quite sufficient and anything more is detrimental. The writer's experience would seem to prove that surface agitation is hardly adequate when caking coals are used. Nevertheless some very excellent results have been obtained. As will be seen from the illustration, the agitator is in the form of a swinging U-tube with the ends hinged to the stationary top of the producer. The walls and the ashpan of the producer revolve and

carry the fire bed around with them. The speed has recently been increased to one revolution in twelve minutes.

19 The ash is removed by a spiral arm lying on the bottom of the ashpan. Ordinarily the arm revolves with the pan so that no ashes are removed. From one to three times every 24 hr. the outer end of the spiral arm is engaged and held fast, thus producing relative motion between the arm and the pan and causing the ashes to flow out rapidly. An ingenious device provides for the automatic release of the ash arm when the producer has made a full revolution. The ashes obtained in this way are exceptionally free from carbon. After the ashes are taken out the fire bed is " broken down " by hand poking and drops from six inches to a foot. The producer is exceptionally well built.

20 In all gas producers that do not provide for continuous removal of the ash there is a definite *cycle of operation,* extending from one ash-removing period to the next. This is usually a 24-hr. cycle, but in some cases it is 12 or even 8 hr., and in the case of the Wood producer it is semi-continuous. At the beginning of the cycle (immediately after the ashes have been removed) the fire bed, though not changed in thickness, is located much lower down in the producer than at the end. This difference in height of the location of the fire bed in the producer from the beginning to the end of the cycle is usually about 2 ft. if the ashes are removed but once in 24 hr., but is correspondingly less when the ashes are removed more frequently. It is therefore apparent that if a producer is to be agitated uniformly it should be provided either with approximately continuous ash removal or with an agitator which automatically varies in height according to the varying height of the fire bed.

21 *The Chapman producer,* Figs. 4 and 5, is the only gas producer sold in parts. The agitator alone may be installed on any stationary producer, or the agitator in combination with the automatic feed may be installed on old or new stationary producers, or, again, a completely mechanical producer including automatic feed, agitator and ash remover may be obtained. The agitator, as will be seen from the illustrations, is in the form of a rake with water-cooled teeth. The rake revolves once in 7 min., and thus every portion of the fire is passed over every 3½ min., which is more than twice as frequent as in any other producer. In most producers agitation depends upon the speed of the walls, which are rather cumbersome to operate rapidly.

22 The agitator is driven by a patent driving head having two long spiral flanges upon which the agitator automatically "screws up" as the fire bed grows higher and "unscrews" when

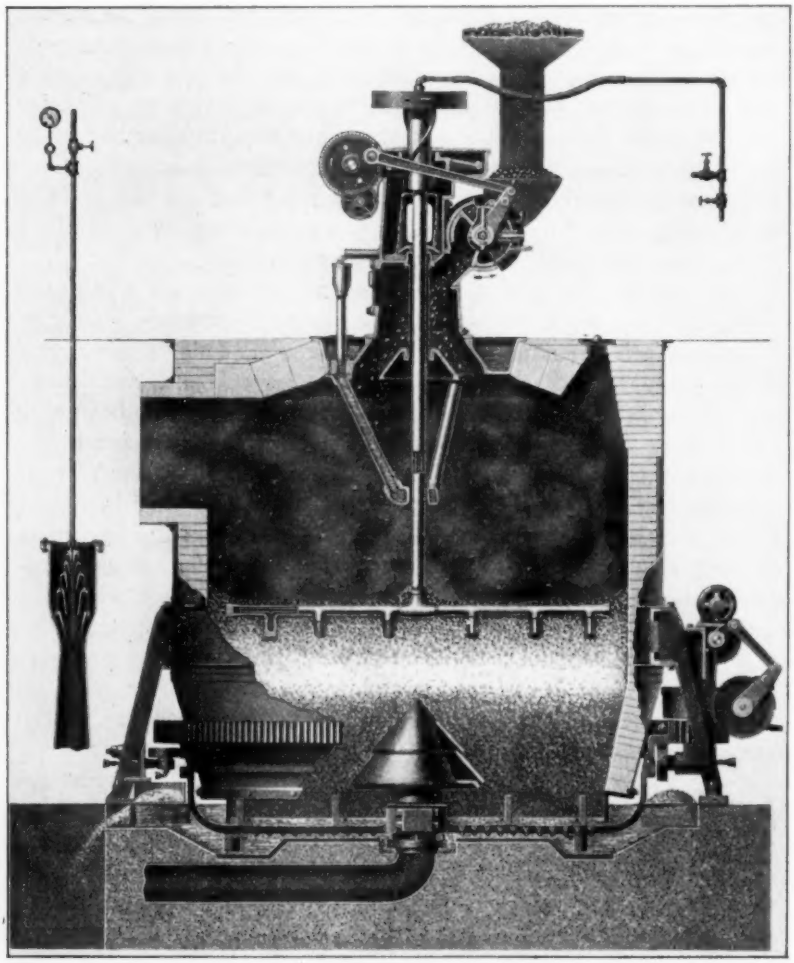


FIG. 5 CHAPMAN CONTINUOUS GAS PRODUCER

the height of the fire bed drops. These spiral flanges are in sliding contact with two revolving lugs which project inwardly from the hub of the driving wheel. As fresh coal is put on the fire it tends

to bury the agitator beneath it. This makes the agitator turn harder through the fire bed, and immediately as the torque is in-

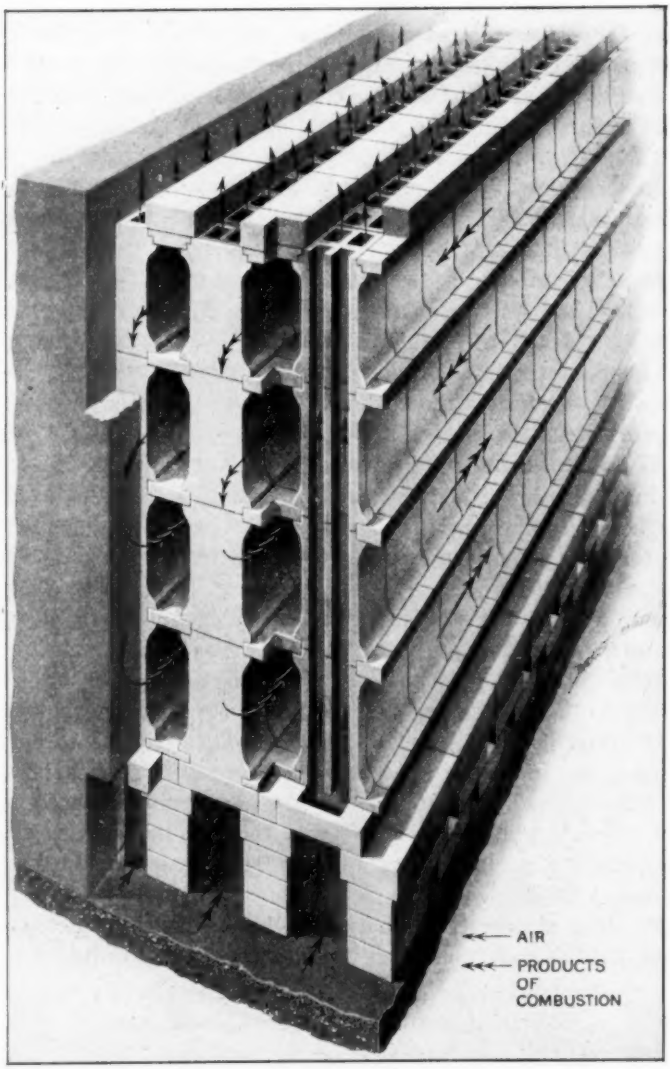


FIG. 6 STEIN RECUPERATOR

creased the driving head screws up to where the forces are again in balance. If the agitator strikes a large clinker fast to the wall

it screws up over it and drops down on the other side, thus avoiding any undue strain on the machinery. The action is a little like that of the Yankee screw driver.

23 The cross-arm of the agitator operates a few inches below the surface of the fire bed, and the fingers project down 8 in. farther. This makes the depth of the agitation about midway between the surface agitation of the Morgan and the deep agitation of the Hughes and Wood. As the fingers project forward the cross-arm immediately fills in the gaps in their wake. The out-stroking effect of the cross-arm tends to pack the fuel against the wall, which helps to prevent blowholes and clinkers.

24 The automatic feed drops the coal evenly over all parts of the fire bed simultaneously, and thus differs from the other mechanical producers in that it does not require a revolving fire bed to spread the coal. The object is to produce slightly better gas-making conditions than when the fuel is dropped only in one corner or sector of the producer at a time.

25 The ash-removing device operates continuously. It consists in a slowly revolving beam extending across the producer through the ashpan and driven by a gear ring which encircles the producer just above the pan, the speed being adjustable to synchronize with the rate the ashes are being made. The speed of the gear ring is adjustable from a revolution an hour to one in ten hours. After the ashes have been forced out by the beam into the outer portion of the ashpan they are picked up by scoops attached to the gear ring and carried to the point of discharge.

26 The Chapman producer makes use of the European idea of agitating the firebed from beneath. The ash beam is provided with fingers which project upward and impart motion both to the ashes through which they move and the fire bed resting on the ashes. This is to help in preventing blowholes and clinkers and to increase the capacity.

27 The size of this producer is 11 ft. inside diameter — the largest made — and the capacity is increased accordingly.

SAVINGS WITH MECHANICAL PRODUCERS

28 With the usual more or less unskilled and indifferent handling, any of the four American mechanical producers can make a gas averaging 150 B.t.u. (low values) instead of the customary 125 B.t.u. obtained in hand-poked producers if operated

with zeal. With skilled handling the best mechanical producers will average 160 to 175 B.t.u., provided the coal is fair and the rate of gasification does not exceed 25 or 30 lb. per sq. ft. per hr., which is fully twice the capacity of hand-operated producers.

29 A Duff producer, operating originally at 10 tons a day and making gas having 125 B.t.u., was fitted with a Chapman agitator and automatic feed, and at 36 tons a day gave an average of 163 B.t.u. Similarly, a Von Kerpley producer, the most popular mechanical type in Europe, gasifying 20 lb. per sq. ft. per hr., making about 135-B.t.u. gas, was equipped with Chapman agitator and automatic feed and changed to 34 lb. gasified per sq. ft. per hr. and 178 B.t.u. This was maintained without difficulty and without any hand poking.

30 The best modern producers properly operated will usually save about 25 per cent of the coal and at least an equal amount of labor. These savings should amount to the total cost of the installation in from one to four years. The Chapman agitator with automatic feed when installed on old hand-operated producers will usually save its cost in the first year of operation.

31 The temperature in a gas producer is highest at the bottom of the fire, and the thicker the fire the cooler the top. The temperature at the bottom should be as high as the fuel will stand without running too much risk of melting the ash. The melting point of the ash in all the good Pittsburgh gas coals is well above 2500 deg. fahr., and the melting point of the ash in Illinois coals is about 300 deg. lower. The temperature at the top of the firebed should be as low as will permit the gas to be conducted to the place of use without forming objectionable tar deposits, and also as low as possible without making the top too sticky and difficult to blow through. Usually a " top temperature " of 1000 to 1100 deg. fahr. is about right, but if the gas is to pass through a water-cooled reversing valve located some distance away 1200 deg. would be better. A " hot top " destroys some of the richest gases and thus wastes fuel. More fuel is wasted in a producer from running with a hot top, i.e., over 1300 deg., than from any other cause.

32 A considerable saving of coal can be brought about by using suitable accessories in the gas house. The most important of these is a pressure regulator, which, when the pressure in the gas main falls, blows the producer harder and makes more gas, and vice versa. A temperature recorder is also of great assistance. A few additional regulating and recording devices are needed if one is to obtain the highest efficiency in daily operation.

THE FURNACE PART OF THE PROBLEM

33 But, *gasifying the fuel is only half the problem of conserving it.* The other half lies in its utilization in the furnace. The two halves of the problem are inseparable. They are both the field of the combustion engineer. Space is too limited to take up more than one kind of furnace — the kind that holds the most promise for fuel conservation and the kind that until the last decade has largely been a failure — the recuperative furnace.

34. A recuperative furnace is never "reversed" and, except in rare instances, only the air is preheated. It costs much less to build and to repair, is easier to operate and gives practically the same efficiency as the expensive and cumbersome regenerative furnace, provided it does not leak.

35 The field of application for the recuperative furnace is very broad and it can be used effectively in both large and small operations, for furnace temperatures as high as 2700 and as low as 1400 deg. About the only uses to which it is not suited are for operations which periodically require a large overload such as the open-hearth process and large forgings over 32 in. in diameter, also for large glass melting tanks. For almost all other purposes the recuperative furnace can be used with great economy.

36 In many operations where the air is not now preheated it will be found that from 20 to 40 per cent of the fuel can be saved by using a good recuperator — one that preheats the air to within 500 deg. fahr. of the temperature of the furnace.

37 The Stein recuperator is shown in Fig. 6 and a typical Stein recuperative furnace in Fig. 7. This furnace is the one most favored in Europe, where, as might be expected, the necessity for fuel economy has caused engineers to give more attention to the problems of combustion than in this country. The recuperator of this furnace preheats the air to within less than 500 deg. fahr. of the temperature of the gases leaving the furnace and is as efficient after six years of operation as when first installed.

38 Another distinctive feature of this furnace is that the gas and air are mixed together a few feet before they enter the furnace but are not given room enough to burn until they reach the combustion chamber. A high heat is then obtained immediately.

39 The recuperative tile in this furnace is set to form horizontal passages for the spent gases, the joints being protected by

a double seal. Each tile is provided with four small vertical passages for the air. These passages form straight chimneys about

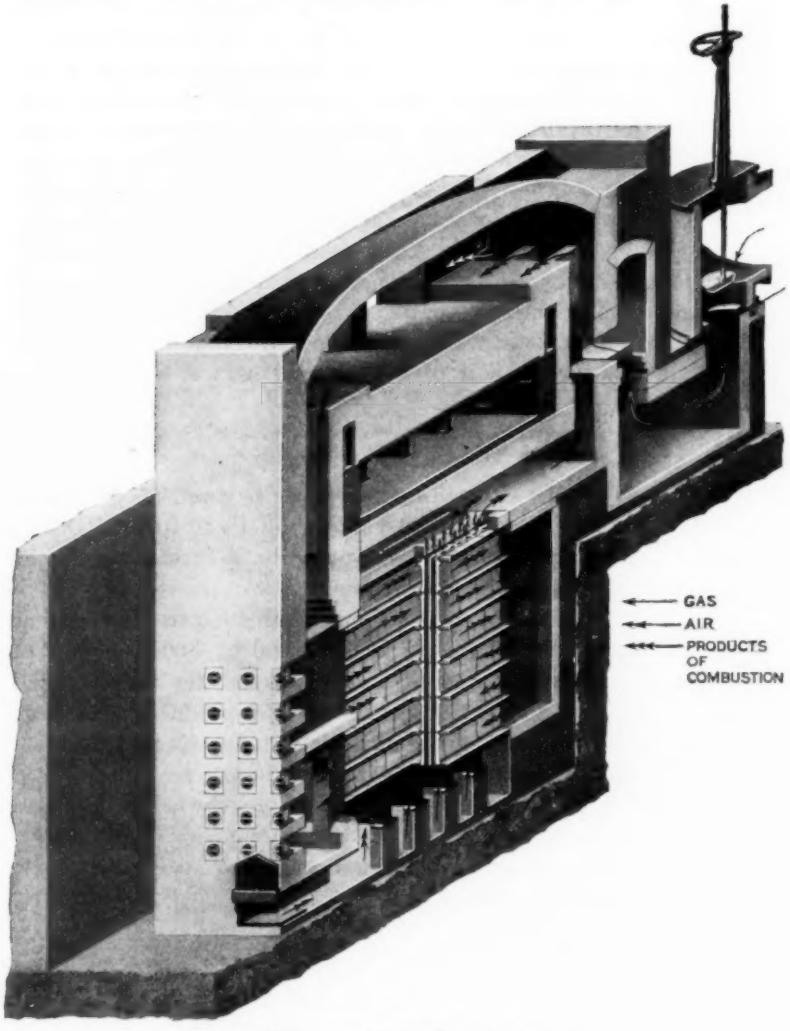


FIG. 7 CHAPMAN-STEIN FORGE FURNACE

6 ft. high with no offsets or turns to hinder the accelerating speed of the rising column of air. The energy represented by velocity of the air thus attained is transformed on entering the furnace into

the energy of pressure. The furnace is thus automatically operated under a slight pressure without requiring the usual blower for the air. The saving of a blower and the power to operate it, however, is not so much of a gain as the fact that the air in the recuperator is not under pressure (from blower) but under suction caused by the chimney effect of the vertical passages, and thus the suction of the air inside the tile balances the suction of the spent gases in the passages between the tile. There is therefore no leakage — in fact no cause for leakage — from the air passages to the gas passages, which has been the bane of the recuperative furnace heretofore.

40 To give one example, a furnace of this description installed in this country nine months ago is now performing a certain heating operation which formerly required 550 gal. of oil a day with 350 gal., a saving of 36 per cent. The saving in oil for the first year will more than pay for the cost of the furnace.

LARGE FUEL SAVINGS POSSIBLE

41 It is not the purpose of this paper to dwell on pulverized coal or oil as fuel, but 20 to 40 per cent of these fuels might be saved if air for combustion were properly preheated from the waste gases by a suitable recuperator.

42 In view of the facts as outlined in this paper we trust that many will agree with us that from 3,000,000 to 4,000,000 tons of bituminous coal could be saved annually by better furnaces and furnace operation, and better producers and producer operation. May we therefore suggest that our Society in the future take suitable action in recognition of the situation herein set forth.

43 In closing the author wishes to acknowledge the kind coöperation of the Morgan, Wood and Hughes companies who have supplied the illustrations of their producers and also some of the information contained herein.

APPENDIX 1

44 As there are more Duff producers in use in the United States than any other type of hand-poked producer, the following test made by the engineers at one of the largest steel works will be of special interest. Former capacity of this producer was 10 tons in 24 hr. and the gas averaged 125 B.t.u. Producer was rebuilt to 10 ft. 6 in. diameter. The capacity of the producer was more than tripled by the addition of the agitator and the change in lining, and the B.t.u. were increased 30 per cent.

TABLE 1 TEST OF DUFF-BRADLEY PRODUCER WITH NEW BLOWERS AND CHAPMAN AUTOMATIC FEED FLOATING AGITATOR

ANALYSIS OF SEVEN-HOUR CONTINUOUS SAMPLE OF GAS

Rate of gasification per 24 hr. = 37.36 tons.
Rate of gasification per hr. per sq. ft. = 36 lb.

CO_2 per cent	Cn H_{2n} per cent	O_2 per cent	CO per cent	CH_4 per cent	H_2 per cent	N_2 per cent	B.t.u.	Steam lb.
4	1.4	0.2	26.8	3.0	10.4	54.2	163	52

Calorific power of gas per min.	552,750	B.t.u.
Sensible heat of gas per min.	94,100	B.t.u.
Total useful heat of gas per min.	646,850	B.t.u.
Average coal fed per hr., lb. by weight	3114	
Average coal fed per hr., lb. calculated from above analysis	3081	
Difference, lb.	33	

EXTERNAL HEAT BALANCE

Dr.	B.t.u.	Per cent	Cr.	B.t.u.	Per cent
B.t.u. per lb. coal	13,039.0	97.5	Calculated power of gas	10,660.0	79.7
B.t.u. steam and air	336.5	2.5	Sensible heat of gas	1,803.0	13.5
			Loss unburnt coal	146.5	1.1
			Other loss (radiation, etc.)	716.0	5.7
	13,375.5	100		13,375.5	100

Total losses, 6.8 per cent.
Efficiency of producer, 93.2 per cent. (counting sensible heat in gas)
Standard gas temperature of 62 deg. fahr. used.
No hand poking was done during test and no clinkers were made.
In calculating B.t.u., U. S. Steel Corpn. lower values are used.
West Virginia coal used carrying 37 per cent volatiles.

APPENDIX 2

45 As there are more Von Kerpley producers in Europe than any other mechanical type, we give the following test. The Kerpley producer, like the other European producers, has continuous ash removal and continuous agitation of the ash bed and lower part of the fire bed, and there is no agitation of the upper part of the fire bed and no automatic feed. Before installing the Chapman automatic feed and floating agitator the capacity was 12 cwt. per hour and in spite of an enormous amount of hand poking the producer was usually full of clinkers. After the new equipment was added the capacity was increased considerably over 50 per cent, no hand poking was required and no clinkers were made. The works engineer estimated that the saving in labor alone was sufficient to pay for the equipment in nine months. The B.t.u. in the gas was increased about 30 per cent.

TABLE 2 TEST MADE IN ENGLAND OF VON KERPLEY PRODUCER, WITH CHAPMAN AGITATOR AND AUTOMATIC FEED

Test on 9 ft. 7 in. inside diameter Von Kerpley gas producer fitted with a Chapman agitator with automatic feed.
Duration of test, 49½ hr.
Average analysis of gas samples taken hourly:

CO_2	O_2	CO	C_2H_4	H_2	CH_4	N_2
4.74	0.30	24.6	0.30	12.80	5.26	52.00

Total combustibles, 42.90 per cent.
Average coal fed per hour, 2408 lb.
Calorific value of gas, 178 B.t.u. per cu. ft.
Net English heat values used: CO—345, H^2—290, CH^4—975, C^2H^4—1590.

HEAT BALANCE

Dr.	B.t.u.	Per cent	Cr.	B.t.u.	Per cent
B.t.u. per lb. coal.....	13,906.0	97.2	Calculated power of gas..	11,498.9	80.34
Sensible heat of coal.....	13.2	0.09	Sensible heat power.....	1,925.0	13.50
Sensible heat of steam	328.8	2.3	Loss in unburnt coal.....	46.3	0.32
Sensible heat of air.....	54.0	0.41	Sensible heat of ash.....	1.8	0.01
			Do. of water passed through agitator.....	55.46	0.38
			Other losses (radiation etc.)	774.54	5.45
	14,302	100		14,302	100

Total losses, 6.16 per cent.
Efficiency of producer, 93.84 per cent.
Producer equipped with Chapman 5-stage blower with $\frac{7}{16}$-in. nozzle.
Coal used: Florence beans, 17.62 per cent, Florence nuts, 30.65 per cent.
Stafford cobbles, 20.73 per cent, Florence cobbles 31.00 per cent.
Coal gasified per sq. ft. per hr. 34.01 lb.
Steam used, 0.278 lb. per lb. of coal.
Gas per lb. of coal, 64.6 cu. ft.

DISCUSSION

EARLE E. ADAMS. We have in this country many small furnaces in which the efficiency is not over 2½ to 5 per cent, and never more than 10 per cent, although efficiencies much higher than this may be obtained when furnaces are intelligently designed and the proper fuel used. It would be almost impossible to use hot or raw gas on these small furnaces.

The author did not say anything about the manufacture of clean producer gas other than to say that he had abandoned the idea of designing a producer that would make clean gas from bituminous coal. We have a producer which makes the gas so that the tar can be extracted, put back into the producer, and gasified. This clean gas is then delivered through small mains to various points where furnaces are located, without fear of clogging such mains or loss due to transmission.

With a recuperative forge furnace similar to the one shown in the author's paper, the typical installation would probably be a number of such furnaces using the equivalent of from 400 to 700 lb. of coal per hour. These furnaces would either be located along one side of a forge shop covering a considerable space, or at various points throughout the plant. It then becomes advantageous to locate the producer at some central point. The logical location is usually alongside of the boiler plant, so that the coal handling apparatus may serve both installations. If raw gas is used in a case of this kind and the producer is operated with an offtake temperature of 1400 deg. fahr., there will be considerable drop in the temperature of the gas by the time it reaches the furnaces, and the loss in sensible heat will average one B.t.u. per degree drop for every pound of coal fired to the producer. If you operate at 1000 deg. fahr. offtake temperature there will be a large accumulation of tar which will be dropped out of the gas and another reduction in efficiency over the normal efficiency at the producer in this type of equipment When a plant is installed with the idea of extracting the tar before it is delivered to the mains, trouble with line plugging, valves sticking, is eliminated, and gas valves may be reversed without difficulty.

There are many advantages in using clean gas in certain classes of work. We who are interested in this type of equipment do not believe it is a " cure all " any more than that raw gas will meet

all conditions, but we do believe that the Fuel Division should give more thought to the use of producer gas and the character of the gas that is to be applied to various kinds of furnace work.

We recall an instance of where city gas at somewhere near $1.00 a thousand cubic feet was replaced by producer gas at a cost equivalent of city gas of about 45 cents a thousand cubic feet. A saving in production cost and the fuel saving in a case of this kind are well worth a great deal of study on the part of the Fuel Division.

J. FRANK ROGERS. The author's interesting and instructive paper presents a subject of importance wherever the conservation of labor and fuel is considered. His presentation should arouse the interest particularly of the executive engineer interested in low production costs.

The economics of producer-gas making is at that stage where the plant executive must look to all possible means for the cutting of costs.

The old day has passed, when the producer was defined (as a manufacturing vice-president once did) as "A thing in which coal was shoveled and gas escaped, a part into the atmosphere and what remained into the gas flues."

The requirements of a gas producer are to make a gas with a maximum calorific value from as cheap a fuel as possible.

To satisfy reasonable demands a machine making producer-gas must meet the following conditions:

1 Make a gas of maximum calorific value delivering in the gas as much of the potential heat of the coal as possible with a minimum allowance for the cost of gasification.

2 Use as cheap a fuel as possible

3 Use a coal as nearly as possible in "mine run" condition, thereby avoiding fine crushing on the one hand and the screening out of the "fines" on the other

4 The complete gasification of the fixed carbon and the resultant ash being free from it

5 The complete mechanicalization of gas-producer operations. The effect of this is to cut down gas-house labor and convert the remaining gas man into a watcher, alert to comply with the varying conditions imposed by the operator of the furnaces in which the gas is used. A parallel to this is the status of a fireman in a boiler

house using oil or natural gas for fuel, who responds to the call of the engine operator for more or less steam by merely adjusting the speed of the liquid or gaseous fuel

6 An elasticity of the producer operation which will quickly pass back and forth from a minimum to a maximum gas output without loss of time or efficiency

7 A low capital investment, and low cost for maintenance

8 A simplicity and ruggedness of design and construction which render wear and tear negligible, and breakdown losses inconsequential, and replacements, when called for, made quickly and if possible without reduction in gas output

9 Elimination of necessity of an extra or spare producer to supply requisite volume of gas in case of a breakdown

10 Independent regulation of air blast and supply of steam needed for gasification of hydrocarbons, and dessication of clinkers

11 Simplicity of operation so that changes of personnel in the gas house do not materially affect gas making in either quality or cost

12 Avoid the use of unusual and expensive materials which are difficult to obtain and expensive to replace in case of loss through breakage

13 A perfectly continuous coal delivery into the producer proportional to the demand for gas rather than an intermittent supply dumped in mass according to the inclination of the operator

14 An open or loose condition of the fuel bed throughout its depth, assisting in the quick liberation of the gas

15 An avoidance of large clinkers and a steady plowing or pushing downward of clinkers in the ash zone and pan, which being of moderate size, render the ash zone porous for the up flow of the air blast

16 A steady mechanical withdrawal of ashes from the producer and their mechanical discharge out of the ash pan.

Mr. Chapman correctly states in his paper, " Much progress has been made in the past 25 years in gas producer construction. It started with the excellent work of W. B. Hughes."

In a continuous production and development of mechanical gas producers during this period, The Wellman-Seaver-Morgan Company of Cleveland have brought the Hughes producer from its initially successful machine up to a machine developed from the experience of the earlier ones in which all of the demands above mentioned are met.

During this career of development and output, nearly 1200 Hughes producers have been built and put into operation in metallurgical, chemical, ceramic, glass, brick, and coke plants.

As in many processes originated and carried forward by the inventive genius of man for the later and more refined improvements, mathematics and technical study have been useful. By their means, the measure of the efficiency of the producer is developd in the "heat balance" of gasification.

In selling producers the salesman has sometimes resorted to the scientifically and practically incorrect standard of considering his gas at a temperature of 32 deg. fahr. Throughout the range of the manufacturing areas of the country, a more nearly normal average working condition is 62 deg. fahr. rather than 32 deg. fahr. Hence, this temperature is now generally accepted in this country as a more suitable base than the unusual 32 deg. fahr. except in some laboratory research work.

In a similar way it is sometimes assumed, though not in this paper, that the entire heat of combustion of hydrogen and its components is available for use in heating the customer's furnaces. As a matter of fact, the waste gases of combustion going out into the atmosphere from the top of the stack contain in a hot and gaseous condition the uncondensed water gas or vapor resulting from the hydrogen combustion. When this is condensed into water as rain drops, the heat of vaporization is released and we have

Heat of hydrogen combustion available for industrial furnaces, B.t.u. per lb.	51892
Heat of vaporization of water vapor condensed in the air, B.t.u.	8740
Total heat of combustion per pound of hydrogen, B.t.u.	60632

While the Table No. 1 in the paper gives the correct value of the gas as 163 B.t.u., according to both of these considerations the corresponding figure in Table No. 2 of 178 B.t.u. uses the English

standard of 32 deg. fahr. and would, when converted into our standard read 165.8 B.t.u.

The unit heat values used are

C_2H_4	1503	B.t.u. per cu. ft, at 62 deg. fahr.
CH_4	909	"
H_2	275	"
CO	319	"

The engineers of the Society are well indebted to the author for directing their attention to the question of producer gas as he has done so well and so entertainingly.

W. W. OAKLEY. This paper presents the case of the mechanical producer extremely well. It also emphasizes the fact that most of the hand-poked producers in this country are " sick."

It has been stated at several meetings of the Society that the overall efficiency of a process is the product of the inherent efficiency of the equipment used times the efficiency of operation. This axiom was never so true as in the case of the gas producer, be it hand-poked, so-called, or mechanical.

The only criterion for judging the efficiency of operation of a producer is the degree of uniformity in which good gas making conditions are maintained in the fire bed. Any device that would automatically maintain the fire bed in a uniform condition continuously would be of very great value. However none of the mechanical producers on the market will do this. All of them require that the side walls be kept free of clinkers by occasional hand poking and some require considerably more manual work. If this is left to the same type of labor and supervision that usually handles hand-poked producers, results will be very similar in either case, not to mention being discouraging. The writer has in mind such a case where some hand-poked producers were converted to semi-automatics but left under the old supervision. Until recently the carbon in the ash was needlessly running 50 per cent. Consequently it is of vital importance that if the expectations of improvement on the installation of mechanical producers are to be realized, the supervision must be intelligent. Also, that if the supervision on hand-poked producers is intelligent and sufficiently vigorous they will closely aproximate the gas quality results of the mechanical producer. As an example, the writer knows three Foster-Miller hand-poked producers from which an average con-

tinuous monthly heat value of 146 B.t.u. per cu. ft., based on the low heating value at 32 deg. fahr. was obtained in regular service. The same organization obtained 151 B.t.u. per cu. ft. from a set of mechanical producers at the same period. The gas analysis in neither case was carried beyond methane which accounts for these figures being somewhat lower than those given in the paper for mechanical producers. However when capacities are considered the mechanical producer as pointed out is far superior to the hand-poked. In the case mentioned the hand-poked producers were gasifying less than 13½ lb. of coal per sq. ft. per hr., its limit, whereas the mechanical group was gasifying 19.6 lb. per hr. about 60 per cent of their capacity. Under these conditions the mechanical producer would show no fuel saving, other than a reduction in the amount of steam used from a ratio of 45 lb. per 100 lb. of coal, to 35 lb. to the hundred. In either case also, good coal must be used for best results.

An additional item that must be prepared for in case of such a change is the larger amount of soot and dust that accumulates with the mechanical producer. This may have a very serious effect if the gas mains are small as the relatively high gas velocities will force the dust into the furnace checkers. From the above discussion it is evident that one of the best reasons for installing mechanical producers is partly to eliminate a difficult labor situation which really controls the fuel consumption as well as a high labor cost. With hand firing under present conditions the labor charges will be close to one dollar per ton fired. With mechanical producers this can be cut to 20 cents per ton fired at the most. This means a saving of over $5000 per year per producer in labor alone.

In the test given in Appendix 1, the writer would like to ask how the gas volume was obtained for use in calculating the calorific power of the gas. Also the term "useful heat" is possibly misleading when sensible heat is included, as there is little agreement among engineers as to the value of the sensible heat of producer gas.

As far as furnace efficiency is concerned we again have the operating factor to contend with. Too often the ideal the operator is working for is a clear stack. This may be attended with from 100 to 200 per cent excess air in a producer-gas fired furnace. Regardless of the perfection of design such a condition will cause disappointing results. In high-temperature furnaces, the items of proper mixture of air and gas and flame control may be of as much

importance as the minimizing of waste heat, when the total cost including repairs of running the furnace per unit of product made throughout its life is considered.

J. H. Taussig. The writer has been a member of the Society for a number of years and does not recall having heard a paper on industrial furnaces and fuels in that time. For many years oil has been cheap, but the time has now come when many industrial plants must burn gaseous fuels. The writer thinks that the fuel division should be divided in order to give proper attention to the important part of the subject. The fact that there are so many industrial fuel furnaces operating at efficiencies of 2.5 to 5 per cent where efficiencies of 30 to 40 per cent might be obtained should lead engineers to give more attention to this subject.

E. N. Trump. The author has not touched upon the subject of the recovery of sulphate of ammonia and other by-products from the coal used in gas producers. The system of recovering by-products is not developed to any extent in this country but has been considerably developed in Europe. At one time the writer built a plant consuming about 100 tons of coal per day in which there was a yield of 80 lb. of sulphate of ammonia per ton of coal. If gas-producer plants were more highly developed they would have a greater economy than has been secured in the past.

Jas. H. Matheson.[1] The author states, "A given amount of time and money, if spent on improving conditions in the gas house, will usually bring larger returns than in any other department. In most industries requiring large heating operations more trouble arises in this department than in any other part of the business."

The great economy gained by the installation of the modern mechanical gas producer over the old stationary, hand-poked and hand-fed producer is undisputed. Installations of mechanically operated gas producers by the Reading Iron Company, Reading, Pa., have reduced the equipment of their old stationary type as above stated practically 50 per cent. The former equipment in one of their departments of 32 of the old type was changed to 17 of the same stationary type, but equipped with the Chapman agitator, thus making the same producer capable of gasifying 1500 lb. of coal per hour per 8-ft. producer, instead of 700 lb. as formerly, with the additional great saving in fuel, labor, and equipment, and the saving of valuable space. The labor saving alone in the first year was about equal to the cost of the agitators.

[1] Reading Iron Co., Reading, Pa.

Although the author deals more directly with the gas producer itself, he does mention the savings made possible by using suitable accessories in the gas house.

One of these accessories, a gas equalizer and soot collector, which is now in successful operation in the Tube Works of the Reading Iron Co., Reading, Pa., is described below.

Very little effective effort has been made to devise means for freeing the soot and dust from the raw gas, between the gas producer and the furnaces, and practically nothing has been done to thoroughly mix and equalize the gas of the several producers.

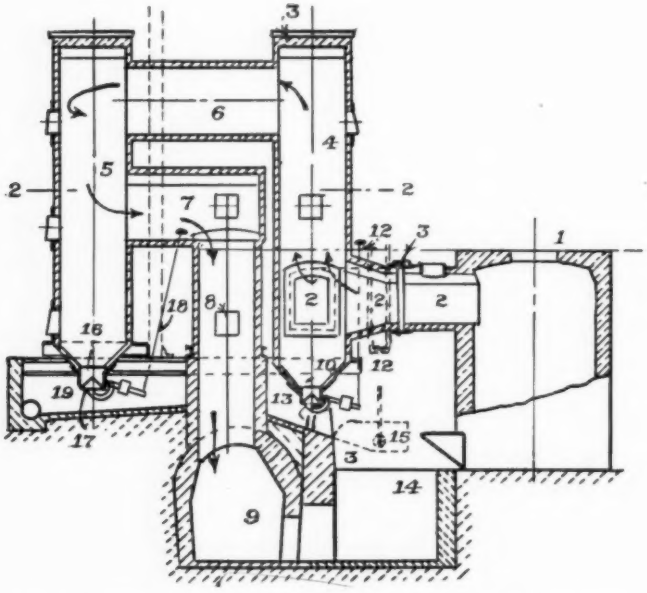


FIG. 8 VERTICAL SECTION, GAS EQUALIZER AND SOOT COLLECTOR

Fig. 8 is a vertical section illustrating the writer's improved gas equalizer and soot collector for gas producers. Fig. 9 is a horizontal section on the line 2–2 of Fig. 8. Fig. 10 is a vertical section on the line 3–3 of Fig. 8.

Referring to the drawings, 1 is a series of gas producers of any of the ordinary types; 2 is a passage leading from each producer. In each passage is a damper 3 of the usual form; 4 is an uptake flue with which the several passages 2 communicate near

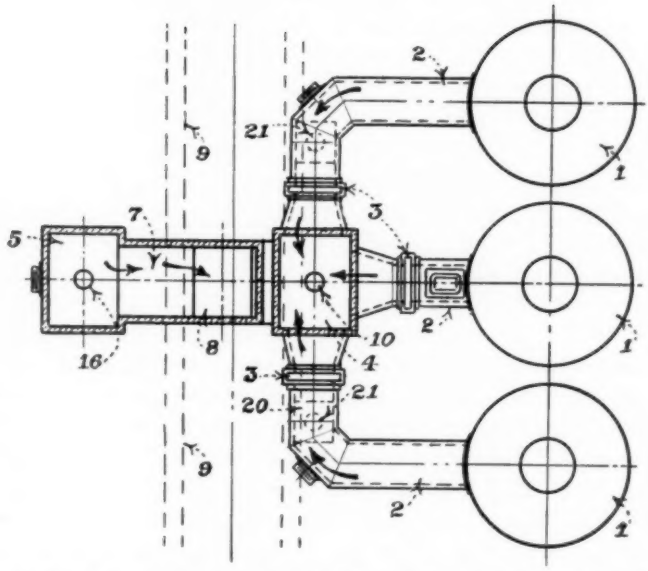


FIG. 9 HORIZONTAL SECTION, GAS EQUALIZER AND SOOT COLLECTOR

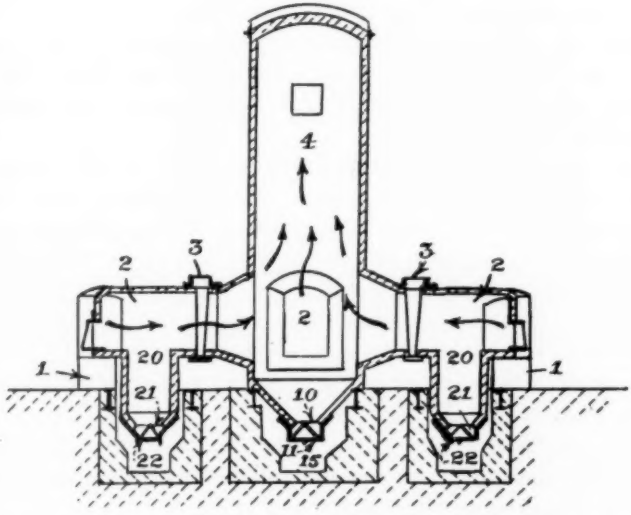


FIG. 10 VERTICAL SECTION, THROUGH 3–3. GAS EQUALIZER AND SOOT COLLECTOR

the bottom, as shown in Fig. 8. The downtake flue is at 5, which connects with the upper end of the uptake flue by means of a horizontal flue 6. Communicating with the downtake flue is a flue 7, which communicates with a vertical flue 8 leading to the main flue 9. In Fig. 8, this flue is shown below the ground level. The lower portion of the uptake flue 4 forms a soot pocket in which is an opening 10 for the discharge of soot. This opening is closed by a valve 11 actuated by a rod 12. The bottom of the flue is preferably tapered towards this opening so that, when the valve is opened, the soot will flow from the bottom of the flue onto the inclined way 13 and into a pit 14. When a traveling bucket 15 is used, as shown by dotted lines in Fig. 8, the bucket can be so located with respect to the inclined way that the soot will flow directly into the bucket. The lower end of the downtake flue 5 forms a pocket in which is an opening 16, similar to the opening in the uptake flue. This opening is closed by a valve 17 operated by a rod 18. The bottom of the flue is beveled towards the opening. When the valve is open, the soot passes through the opening 16 and collects on a floor 19. Each side passage 2 has a soot pocket 20 in the present instance. The bottom of each pocket is beveled towards an opening 21. Each opening is closed by a hand-operated valve 22, similar in construction to the valve 11.

The uptake flue 4 and the downtake flue 5 are preferably rectangular in cross section to provide broad and flat surfaces against which the gas impinges as it passes through the flues, thus more readily separating the soot and dust from the gas and thoroughly mixing the gas.

By the above construction, the gas, as it travels through the producers and through the several flues, separates from the soot and dust which falls into the pockets. When sufficient soot has collected in the pockets, the valves are opened so as to discharge the soot and dust from the pockets. By having the extended uptake flue 4 and the extended downtake flue 5, the separation of the soot from the gas is assured, and the gas, when it reaches the main flue 9, is comparatively free of soot so that the cleaning of the main flue is obviated.

By constructing the soot collector in the manner described, the gas of the several producers is averaged and provides a uniform quality of gas at all times.

While a unit of three producers is shown the principle is applicable to any number.

The following are a few advantages of the above installation:

Saving of fuel
Increased product
Improved quality of product
Saving in gas valves
Less brick work and labor
Less machine shop repairs
Less labor cleaning out soot during and end of each week
More continuous operation of furnaces
More contented workmen.

In addition to the above the soot recovered is found to have considerable value, and preparations are now being made to briquette same and possibly burn under boilers or other furnaces or it can be used for household purposes.

The analysis of the soot and dust is as follows:

Sulphur	1.00
Moisture	0.77
Volatile combustible matter	0.18
Fixed carbon	78.71
Ash	17.34
B.t.u. per lb.	12184

The soot and dust recovered from one stationary producer 8 ft. inside diameter and 14 ft. high, equipped with a Chapman mechanical agitator gasifying 1500 lb. of Westmoreland gas coal per hour, is approximately, 4600 lb. per week of 5½ 24-hour days, about 2 per cent of the total coal gasified, or in volume over 9 cu. yd.

Particular attention is drawn to the small diameter of the Reading Iron Company producers, and the large increase in gasification, after installing the Chapman agitators. By urging the producers, we can increase the gasification to 2000 lb. of coal per hour, but do not advise it.

We have found it is possible by the use of the gas equalizer and soot collector to eliminate one producer out of every four. This will apply to any of the mechanically operated producers noted by the author. This installation will also make possible the use of a cheaper grade of coal. It will be noted by the illustrations that there is no complication, and the installation is not ex-

pensive. The application to a battery of producers of three or more can be made very quickly.

The Reading Iron Company are equipping their entire producer gas plants with the above improvement, resulting in a clean gas without washing, retaining the sensible heat of the gas in addition to the B.t.u., value of the tar and volatile hydrocarbon.

THE AUTHOR in answer to Mr. Trump wrote that there have been about six serious attempts in this country to install and operate a by-product gas producer plant using the method invented and practised in Europe by Ludwig Mond and his followers. All these attempts have been signal failures. None of the plants are now in operation although the last was installed but three years ago at a cost of over half a million dollars.

The writer has talked with those who have had their money invested in five of these attempts and they all agree that the plants are excessively expensive to build and also to operate and that the yield of ammonium sulphate is not as large "as advertized." Some day something simpler and better than the Mond process will be evolved. We hear of eighty and even ninety pounds of sulphate of ammonia per ton of coal but the plants the author has visited in Europe averaged nearer sixty-five pounds. In one of the largest plants in England during the war when the coal was poor the average yield was 40 to 45 pounds. In South Staffordshire the writer was told that that much heralded plant had not been able to pay dividends; in fact they had given up trying to use the coal from the mine at whose mouth the plant was located and they were buying the coal from a mine twenty miles away in order to get a living yield of sulphate. In the author's opinion we must look to some other solution of the by-product producer than that devised by Ludwig Mond or his followers.

In reply to Mr. Adams: Raw producer gas owing to the tar and soot it contains cannot easily be applied to a number of small furnaces located some considerable distance from each other. When the gas is washed the first cost of the installation is about double the cost of raw producer gas and the added expense of operation makes the cost of clean gas per available heat unit about double that of raw gas. But in spite of this doubling of cost clean producer gas is usually only half as expensive as city gas and has a legitimate field of use midway between raw producer and city gas. Some day it may be possible to make clean producer gas at a cost not more than 50 per cent in excess of raw gas. It will then be much more popular, provided the equipment is easy to operate.

Mr. Oakley emphasizes the personal factor in operating gas producers and furnaces. The author would go still further and say that gas producers and furnaces should be classed or valued according to the ease or difficulty of maintaining constantly in them good working conditions. The producer in the English test given has been in operation over a year and has only been poked by hand Sunday nights when starting up after the shut-down of Saturday afternoon and Sunday. And in the furnace described the air is seldom regulated more than once a week provided the load is steady.

Mr. Oakley says that it is possible to make good gas in a hand-operated producer provided it is poked with sufficient zeal and intelligence. But who wants to do it when an agitator will do more "intelligent" poking than three men on a shift and three shifts a day?

In answer to the question as to how we figure the quantity of gas made per pound of coal we start with the ultimate analysis of the coal and from it figure the amount of carbon in a pound of coal; and from this we deduct the amount of carbon lost in the ashes and in the soot. We then assume that the remaining carbon has to be accounted for in the gas. From the volumetric analysis of the gas we figure the total carbon content of 1 cu. ft. using the atomic and molecular weights and percentages of all of the carbon containing gases present. Then by dividing the weight of carbon passing into the gas from one pound of coal by the weight of carbon appearing in one cu. ft. of gas we obtain the number of cu. ft. of gas made per pound of coal.

In regard to using the term "useful heat" instead of "sensible heat," ordinarily it would be more correct to say "sensible heat" but if a recuperative furnace is employed all sensible heat is useful.

Mr. Matheson's comments on handling the large quantity of soot made in mechanical producers operating at a high capacity are interesting. The author has seen his gas equalizer and soot collector in operation and it is very effective. All mechanical producers should be supplied with a complete soot collecting system. The quantity of soot he mentions, however, is excessive. The author was once called in to a plant where their 6-ft. inside diameter gas flue accumulated 3 feet of soot in 3 days. We changed from Pocahontas coal to a cheaper grade and from a hot top temperature to a cold top. As a result the flues would run a week with less than 3 feet of soot and the trouble was overcome. Manufacturers

of gas producers realize that in the future both they and their customers must give more study to the question of soot prevention and soot removal. Excessive soot in the flues causes excessive back pressure which in turn is apt to result in a reduction of both the quality and quantity of the gas made. If more users of producer gas would follow Mr. Matheson's example there would be less trouble and much coal would be saved.

In closing the author wishes to state that the trend of present practice both with furnaces and producers is toward simplicity — more simple and durable construction and more simple operation.

No. 1815

AVOIDABLE WASTE IN THE OPERATION OF LOCOMOTIVES AND CARS

By William Elmer, Altoona, Pa.
Member of the Society

The average freight locomotive travels less than 60 miles per day. It spends about half of its time either in the hands of the transportation department moving trains or ready to move them, or in the hands of the motive-power department being repaired and prepared. There is naturally avoidable waste in each department, and questions which may accordingly be asked relate to whether the engines are properly loaded and properly used. The present paper outlines the procedure for determining this, and one of its several appendices gives a method of working out the most economical tonnage for loading the freight engines of any division, based on actual practicable performance in everyday operation. The treatment considers the value of the locomotive, taking account of interest, depreciation and taxes; the relationship between straight-time and overtime rates for road crews; the quickening up of the time of the trains by a reduction of tonnage and the increase of the time the crews are on duty by an increase in tonnage. When these matters have been sufficiently studied in the light of the recorded facts, the question relating to proper loading can be intelligently answered.

The author discusses avoidable waste in the operation of cars under three heads: (a) Their utilization in the hands of agents, shippers and consignees; (b) their handling and dispatchment in yards and on the road; and (c) their repair and inspection.

Things contributing most to the reduction of waste in car and locomotive operation are coöperation and teamwork. If a division superintendent can be assured that everything is being done that can be done to have every available engine in service that can be put in service and that every engine dispatched is being loaded to the maximum number of cars it can economically haul, then he is assured of an economical performance and an avoidance of waste in the operation of both locomotives and cars.

LOCOMOTIVES, the first of the two divisions set forth in the title, are classified into major groups as freight, passenger, shifting and work locomotives. There are 65,000 locomotives on the railroads of the United States and half of them are in freight-train service. Thirty-two thousand and eighty locomotives earned a

Presented at the Annual Meeting, December 1921, of The American Society of Mechanical Engineers.

freight revenue of $4,325,078,866 in 1920, or an average of $135,000 per locomotive per year. Each engine made an average of 59.3 miles per day or 1800 miles per month. The average freight engine earned for its owners $370 per day or $6.25 per mile run. This is at the rate of $15.40 per hour or about 26 cents per minute. The striking thing in the group of facts above presented is the figure of 59.3 miles per day made by the average freight locomotive. How can we excuse an average mileage for all the freight locomotives in this country of less than 60 miles per day? We can picture the average freight locomotive rolling along the rails at 15 miles per hour and that means less than four hours out of each twenty-four actually moving trains. The locomotive spends its entire time either in the hands of the Transportation Department moving trains or ready to move trains, or in the hands of the Motive Power Department being repaired and prepared. Roughly we may say that the engine is in the hands of each of these departments about half the time. Of course there is avoidable waste in each.

2 Taking up first the Transportation Department, there are two broad inquiries which may be made: — (*a*) Are the engines properly loaded? (*b*) Are they properly used? Assuming that suitable engines have been furnished the Transportation Department, or taking the engines on any division as we find them, how are we to know when they are properly loaded? If a dynamometer car is available, road tests may be run to determine the drawbar pull of the engines and to measure the resistance of trains of various make-ups on the ruling grades at the desired speeds. In the absence of this facility it may be desirable to outline the procedure.

A — ARE THE ENGINES PROPERLY LOADED?

3 A track chart of the road is necessary, giving the distances from the starting point to the beginning and ending of each curve and tangent, with the degree of curve, and elevations of points where the grade changes. With these data a true profile may be plotted, showing the elevations above sea level and the actual grades; but this profile will not be fully representative of the resistances encountered by moving trains until it has been transformed into an equivalent compensated profile by superimposing the curve resistances on top of the grade resistances for each direction of traffic. We can imagine a level railroad so full of sharp curves that a very considerable resistance would be experienced by a moving train. Many experiments have been tried in an effort to find how much

resistance various curves offer to a moving car, and we will take 1 lb. per ton of 2000 lb. per degree of curve. The resistance due to grade is fortunately an exact mathematical quantity — 20 lb. per ton for each 1 per cent of grade. Therefore each degree of curve offers the same resistance as a 0.05 per cent grade. A 6-deg. curve had the same resistance as a 0.3 per cent ascending grade. A grade which is climbing upward at the rate of 26.4 ft. per mile or 0.5 per cent and has in it a 6-deg. curve, or 955-ft. radius, will therefore have superimposed on the true grade of 0.5 per cent the equivalent resistance of a 0.3 per cent grade due to the 6-deg. curve, or a total equivalent grade of 0.8 per cent. Of course, to a train coming down this hill, the equivalent grade would be the difference between these values, or 0.2 per cent. To those who are interested in the details of the methods to be followed in constructing equivalent profiles, where a road has not been built with compensated curves, reference is made to Appendix No. 1.

4 Having determined the equivalent grade, it will be necessary to decide whether it can be operated as a momentum grade or not. If the length of the grade or other physical conditions on the approach prevent attaining any considerable speed, the dead pull of the locomotive will have to be depended on to get the train over. The tractive power of a locomotive is readily calculated from a very simple formula where p is the boiler pressure in pounds per square inch by gage, d the diameter of cylinders, l the length of stroke and D the diameter of the driving wheels, all in inches. For a simple two-cylinder engine, tractive power $= 0.85pd^2l/D$. When a locomotive is moving, some of its tractive-power effort is used to overcome friction of the engine and tender, and on a grade some more is needed to lift its weight against gravity, and at speeds of more than six or eight miles per hour the boiler becomes a factor in its inability to furnish enough steam to follow the pistons with full pressure under long cut-off conditions, so that some more complicated formula becomes necessary in the calculation of the tractive power required for moving trains. Besides the resistances due to curves and grades, trains are affected by journal and flange friction, wind, rolling resistance, temperature, etc. A formula for locomotive tractive power and train resistance is presented in Appendix No. 2. It is useful for speeds above 10 or 15 miles per hour.

5 It is a well-known fact that trains cannot be loaded on tonnage alone. One hundred empty cars weighing 20 tons each would be a 2000-ton train, and might overload an engine to the

stalling point, whereas the same engine on the same grade would handle twenty-five 80-ton cars with no trouble. The number of axles is the important factor, and in order that a long empty train may have the same resistance as a short loaded train, it is necessary to use a factor for each car known as the adjustment factor and this factor will vary with the different physical conditions met with on different divisions. A discussion of adjusted tonnage rating will be found in Appendix No. 3.

6 Having discovered the adjustment factor for any given division, and knowing the principal types of freight engines in use on that division, it is well to construct tractive power-speed curves for the various engines, and plot on the same sheet adjusted-tonnage train-resistance curves on various level and compensated-grade tracks, so that the intersection of the tractive power curve with any given grade will show the speed that could be maintained with a full-tonnage train on that grade. A sample curve sheet with the method of calculation will be found in Appendix No. 4.

7 After having completed the above described investigations and having before us the equivalent profiles and the speed curves on various grades, we can lay out a schedule of the running time between the various towers, adding the necessary time to cover the initial and final terminal delay, water stops, coal and fire-cleaning stations, interference from passenger trains, etc., and bearing in mind the overtime limit based on a speed of $12\frac{1}{2}$ miles per hour for the distance between terminals and the time the crew is on duty. The preparation of a freight-train time table will be found in Appendix No. 5.

8 Now comes the crux of the whole matter. After the tonnage has been established, what are the results on the road? Do the trains lose so much time sponging or setting off cars with hot boxes, or draw heads out or brake rigging down, or due to interference from other trains that they cannot get over the road without excessive overtime? If the dispatching and terminal and road supervision are all that they should be and a record has been made for a sufficient period from which may be drawn reliable conclusions, we can determine whether the overtime is excessive — in which event the tonnage should be decreased — or if the majority of the trains get over the road within the overtime limit, then the tonnage should be increased. Appendix No. 6 contains the description of a graphical report showing each morning the performance of each of the previous day's trains, both slow and fast freight, plotting the time on duty

from called to relieved against the percentage of the full tonnage loading of the engine utilized. This gives the train master, road foreman of engines and superintendent a review of the preceding day's operations, and any falling away from the standards set up on the part of the subordinate officials whose duty it is properly to load the trains is quickly brought to light. In Appendix No. 7 is given the method of working out the most economical tonnage for loading the freight engines of any division, based on actual practicable performance in everyday operation. The treatment considers the value of the locomotive, taking account of interest, depreciation and taxes; the relationship between straight-time and overtime rates for road crews; the quickening up of the time of the trains by a reduction of tonnage and the increase of the time the crews are on duty by an increase in tonnage. When these matters have been sufficiently studied in the light of the recorded facts, we are in a position to answer the question, Are the engines properly loaded?

B — ARE THE ENGINES PROPERLY USED?

9 So far as the Motive Power Department is concerned it is important to have reliable reports which present promptly to the responsible operating officers, on the succeeding day if possible, all the pertinent facts concerning the performance of the locomotives available. These reports should cover not only the utilization made of the serviceable locomotives but also of all those laid off for repairs, both in the roundhouses and the back shops. The more promptly the work is done the more engines will be available for service and the smaller will be the number required to be purchased and to bear interest and depreciation charges. To this end the facilities at the engine terminals should be ample to inspect promptly the incoming locomotives and send the reports to the dispatcher, who can at once call a crew in case the engine has only light work which can be completed by the time the crew reports. The fire-cleaning pits and facilities for handling ashes, coal, sand and water should be in duplicate at important points, and at one well-known freight-engine terminal it is possible to clean the fires and prepare for service 400 locomotives per day. Hot-water systems for washing and filling boilers save time, and drop tables or unwheeling hoists should be provided for handling driving wheels, spring rigging and driving-box repairs. Ample jib or overhead cranes should be installed in all important enginehouses, as the rods, pumps, pistons, smokebox

fronts, etc., of modern locomotives are now so heavy that mechanical appliances must be used to reduce the cost of handling and save time in running repairs. The enginehouse referred to above at times furnishes the power for ten eastbound trains in two hours and at the same time ten to fifteen engines an hour for westbound trains. An operation of this magnitude requires close supervision in order to avoid waste of power and loss in efficiency.

Cars

10 The avoidable waste in the operation of cars may be considered under three heads:

a The utilization of cars in the hands of agents, shippers and consignees

b The handling and dispatchment of cars in yards and on the road

c The inspection and repair of cars by the maintenance of equipment department.

A — THE UTILIZATION OF CARS IN THE HANDS OF AGENTS, SHIPPERS AND CONSIGNEES

11 The committee of engineers appointed by Secretary Hoover some months ago to investigate waste in industry made a most amazing report. They undoubtedly gave this question careful study and the report that they made brought out the fact that the production of this country could be immediately increased about 50 per cent by the full utilization of existing facilities. The management of the industries of this country are acknowledged to be the most efficient known and the railroad managements are no exception, therefore it is the more surprising to find such disparity between the present efficiency and the attainable.

12 So far as the railroads are concerned, one great means that suggests itself is the increased use of cars in the hands of agents and shippers, which necessarily involves the promptness with which they are loaded and unloaded and the extent to which they are loaded, i.e., that the maximum loading be secured for the car in the minimum time, etc.

13 Maximum car loading is a matter of dire necessity during periods of car shortage. It is also very essential to the economic conduct of transportation. During recent months the necessity

for conserving cars has been decreased by the small volume of tonnage handled by the railroads; the requirement of economy is more urgent than at any time within the past eighteen months. This is all a matter of education and a spirit of coöperation on the part of the public. When the railroads had more business than they could handle the local officers were being urged to get out and interview the shippers, the underlying idea being to secure their coöperation, especially in the matter of better and heavier carloads. Distinct improvements were noticeable and the effort proved its worth. With the slump in traffic the "drive" lost its punch and there is a tendency on the part of the shippers and railroad men alike to let down in their efforts to secure maximum loading. This "line of least resistance" method is resulting in considerable less than capacity loading.

14 It is a fact, not generally recognized, that car loading affects the cost of railroad operation very seriously, not only because the paying load may be a small percentage of the gross train load, but also because lightly loaded cars require more tractive effort per ton than heavily loaded cars, e.g., the average weight of a car is from 15 to 20 tons while the average weight of all commodities is averaging approximately 27 tons. The load of the car itself must be hauled with every movement of the contents and requires as much tractive effort on the part of the locomotive per ton to move this weight as it does for the contents, therefore the importance of keeping the percentage of lading to total weight as high as possible is self-evident. This question has assumed a very different aspect to the shipper since the passage of the Transportation Act, which stipulates that the rates must be sufficient to earn a fixed return on the value of the properties. Any waste due to the light loading of cars adds to the operating cost and thereby to the rates necessary to earn the specified return. The shipper therefore has a new interest in effecting economies of transportation and can contribute to that end most effectively by coöperating in the heavier loading of cars.

15 During period February to August, 1920, the average loading of cars in the United States increased from 28.3 tons to 29.8 tons per car. As a result of this coöperation on the part of shippers there was a gain of carrying capacity equal to approximately 112,500 cars. From time to time we have noticed by the public press that there is a shortage of 100,000 cars in the United States, and as such is considered a serious matter to the trade of the country, it is very

apparent that the simple feature of increasing the load in each car 1.5 tons more than liquidated the alleged shortage.

16 General practice permits the loading of cars 10 per cent in excess of the marked capacity. There are great possibilities in the utilization of this margin for with many classes of loading great advantage may be taken of it to gain one car in every ten and to increase the average carload correspondingly.

17 There are many commodities moving which will permit of the making of trade units to correspond to the capacity of the car; this has been done with cement, and other like commodities. The trade unit for cement shipments was set at 144 bbl. for a 50,000-lb. capacity car; 173 bbl. for a 60,000-lb. capacity car; 231 bbl. for an 80,000-lb. capacity car and 289 bbl. for a 100,000-lb. capacity car. The establishment of this standard encourages the loading of cars to capacity. If this were done with flour and all similar commodities great assistance might be rendered to the railroads.

18 The agent through close association with shippers is in the best position to encourage maximum loading. It is often decidedly hard to convince shippers that they are not loading their cars to cubical capacity. This is particularly true of the coal operators. The best means of producing convincing evidence of the empty space in a car is to show the shipper a photograph of the car which will speak for itself, and we have found a kodak to be a most helpful instrument in increasing the tons per car. The cars may be intercepted and photographed at scales or in classification yards.

19 The prompt release of cars under load is a large factor in the efficiency of the car. Most shippers and consignees are reasonable in this respect and will give us their best efforts if the matter is handled with them in a diplomatic way. After urging the shippers and consignees, the railroad then has a very important part to play by the prompt movement of cars, whether loaded or empty; it being purely psychological that after urging the shipper or consignee and then failing on our own part would necessarily breed antagonism.

B — THE HANDLING AND DISPATCHMENT OF CARS IN YARDS AND ON THE ROAD

20 After cars have been loaded and waybills furnished by the agent to transport freight from point of origin to destination, it becomes the duty of the train master to arrange for movement and delivery with the least possible delay consistent with economical

operation. This necessarily involves good organization and effective supervision to accomplish proper movement through yards and over the road. Normally cars are weighed at the first track scale encountered after departure from shipping point, for the purpose of ascertaining proper charges for transporting freight which involves the agent at the scales, who is responsible for securing accurate weights as a basis for applying the freight charges. As a rule, the weighing is performed in a yard consisting of receiving and classification tracks. Trains arriving in the receiving yard are subjected to inspection and minor repairs to insure safe movement over the road between terminals, sending to the repair yard any bad-order cars that must be "shopped" for this purpose. After inspection and repairs have been completed, the train is prepared for switching from receiving yard to classification yard, which process requires car markers to chalk-mark cars for their respective classification tracks, according to destination and routing, also furnishing corresponding switching lists for the conductor in charge of switching crew, and switchmen who operate switches leading into the classification yard. Yard locomotives and a force of trainmen are required to switch trains into the classification yard at proper speed for accurate weighing at points where cars pass over a track scale; also requiring brakemen, commonly designated as "car droppers," to ride cars into the classification yard and control them by use of hand brakes to bring them to a stop at the proper point and to avoid damage by impact with preceding cars standing on track.

21 The train from receiving yard has now been distributed on various tracks in the classification yard, usually from ten to thirty tracks, depending on the size and importance of the yard operation. The original train having thus lost its identity, following trains, classified in like manner to the same tracks, are required to assemble cars that will comprise new trains to be dispatched when the required tonnage is accumulated. A variety of conditions arise at this stage of the operation that seriously influence the time consumed by cars enroute to their destination, which may necessarily be repeated from one to many times between the originating point and destination of cars, depending on the distance and the territory over which they are moving. The time required to assemble sufficient tonnage for a train in the classification yard is very largely dependent on the steady or intermittent arrival of trains in the receiving yard; also on the hauling capacity of road locomotives used on trains dispatched in the same direction, which may be 35

or 50 cars from one yard and 100 to 115 cars from another yard for the same class of locomotive, depending on the ruling grade of the division over which trains are being hauled.

22 In this connection another primary cause of delay in assembling trains in the classification yard is to be found in the usual number of classifications imposed upon certain yards for the convenience of connecting divisions to meet their requirements for various reasons, but primarily due to inadequate track and switching facilities. So-called "prior classifications" are also a source of yard delay at the point where they are assembled, but the time thus consumed is presumably offset by saving in time at the next yard or terminal point where such trains are kept intact and delivered to the division in advance thereof without reclassifying, which means an actual saving in the aggregate time consumed from shipping point to destination, also in operating expenses. Therefore a considerable portion of yard delay is beyond control, owing to prevailing conditions that cannot be eliminated. However, there is ample opportunity for minimizing yard and road delays to train movement by employing the best operating methods, maintaining sound organization and efficient supervision.

23 Time consumed assembling tonnage for heavy trains to be hauled by large types of locomotives over comparatively level grades may be viewed by some as a contributory cause of "avoidable waste in cars," but it should be recognized that doing so reduces the number of engine and train crews and locomotives required to haul a large volume of freight, which means economical operation.

C — THE INSPECTION AND REPAIR OF CARS BY THE MAINTENANCE OF EQUIPMENT DEPARTMENT

24 When trains are hauled over the road certain defects develop, and by the time they reach the terminal of a run a certain portion of the cars, say 3 to 5 per cent, must go to shop for repairs. The cars from the time last inspected until they go to the industries or mines to be loaded and return, develop many defects due to stress and strain of service, and consequently they have to be shopped when reaching the nearest terminal.

25 There are three distinct classifications for shop cars: light, medium and heavy. When an inspector discovers a defect which he cannot correct, he applies a bad-order card, and places it in a conspicuous location on the car. These cards are placed in one of

three positions: vertical, horizontal or at an angle of 45 deg. A card placed in a vertical position indicates light repairs; horizontal, heavy repairs; and at a 45-deg. angle, medium repairs.

26 After the train is inspected a car marker goes along the train and marks each car with chalk and the date and track number to which each particular car is to be classified.

27 As already stated, the classification yard consists of a number of tracks for the various classifications; a certain number of these tracks are set aside for shop cars, some tracks for light-repair cars, some for medium, and some for heavy repairs. The shop cars are moved from the classification yard to the car repair yard by shifting engines, usually at night, and the repair tracks filled up so that when the gang foremen arrive at their places of duty approximately one-half hour in advance of the commencing time for the shop forces, they can prepare their work for the day. It is their duty to see that the work is properly distributed and there is sufficient material on hand to proceed with the repairs.

28 After they have all the cars written up they start inspecting the work done by the men to determine if it has been properly performed, and that proper charges for material have been made. After this is completed they report out such cars as are ready to go and mark those uncompleted to be set back. The shifting crew then takes out the entire string, returning those not completed. It is the gang foreman's duty to see that any cars not completed are in such shape that they can be moved out with the O.K. cars, as otherwise there would be non-movable cars between two or three cars O.K. for service and this would hold the cars ready for movement an unreasonable length of time.

29 After the cars are turned out of the repair shop they are returned to the receiving yard and reclassified over the hump to be returned to service.

30 In order to keep a check on cars undergoing repairs, a report is prepared and sent to the different operating officials. The report indicates the time the car is shopped, the time it is moved to the shop, the time repairs are completed and the time car is moved out of shop. By checking over this report each morning the master mechanic, superintendent or general officers can determine in a few seconds if there are any cars that are being held an unreasonable length of time.

31 It is the aim of the operating officials not only to see that all cars are repaired, but to have the cars repaired promptly and returned to service in the most expeditious manner.

CONCLUSION

32 The great secret of the entire operation, therefore, is coöperation and teamwork, and these can be checked by suitable reports.

33 The statistics which reach the superintendent's desk giving hourly, daily, weekly and monthly information are many and varied, and originate from numerous sources, but the reports scanned by the author with most interest each day are those which tell where each of the heavy road freight and passenger engines were the day before and what they were doing. There is a maxim, "Take care of the shillings and the pounds will take care of themselves." It seems to apply particularly to the railroads. Take care of the engines and the dividends will take care of themselves. Of course this could not be literally true, but there is so much involved in this "taking care of the engines," embracing as it does the time and inferentially the money spent in locomotive repairs, the quality of back-shop and enginehouse work performed, the proper tonnage rating, and suitable loading of engines in order to obtain the most economical road speed, the reduction of delays getting into and out of yards, the inspection and repair of car equipment, the efficiency of water stations, coal-, sand- and ash-handling plants, the organization and operation of wreck forces, the handling of local freight and work trains, in fact, almost each and every one of the thousand and one matters that go to make up a successful operation of a division. If any one of the features named above is not functioning properly, as well as others too numerous to mention, the effect will be seen in the slowing down of the road speed or a lowering of the average mileage per serviceable locomotive or a falling off in the loading efficiency. All these must be at their highest possible levels of practical performance, and when they are, a glance of the eye at the daily barometer ought to tell it, and when they are not, a few minutes' inspection of the data ought to tell why and point the remedy. The supervisors must have tracks fit for speed and service; the signal engineer must have communicating systems and signal apparatus in good working order; the road foreman must have engines properly rated and sufficient crews and supervision; the train master must have his yard and road forces properly instructed and disciplined; the division operator must have his train dispatchers and signalmen alert and intelligent; and the master mechanic must produce the power in ample quantity and fit for

service. If the division superintendent can be assured that everything is being done that can be done to have every available engine in service that can be put in service, and that every engine dispatched is being loaded to the maximum number of cars it can economically haul, then he is assured of an economical performance and an avoidance of waste in the operation of locomotives and cars.

APPENDIX NO. 1

EQUIVALENT PROFILES

34 Fig. 1 shows a portion of a railroad between Block Stations QY and FR. At the bottom is the track chart which locates the curves and tangents, mile posts, stations, etc. The distances from the starting point are given in feet and above is shown in solid line the true profile as plotted from the elevations given in Table 1, column 3.

35 Many railroads will have the information shown in columns 1, 2, 3, 7 and 8 without making any special survey, and from these data the other columns of Table 1 may be compiled. We observe from the first line that Block Station QY in column 1 is the 0 or starting point with an elevation above sea level, of 803.12 ft. It is on a 2-deg. curve as shown in column 8. Line 2 shows that at 292 ft. from the starting point this 2-deg. curve stops, and the 292 ft. times 2 deg. equals 584 as entered in column 9. Multiplying by 0.05 and dividing by 100 gives an equivalent grade rise of +0.292 ft., column 10.

36 As there is no actual top of rail elevation recorded at this point, it will be necessary to interpolate, and line 3 gives an elevation of 806.12 at 934 ft. from 0. The difference of 3 ft. is recorded in column 5 and dividing by the distance, 934 ft., column 4, gives a grade of 0.321 per cent, column 6. Consequently 292 ft. of this grade would result in a rise of 0.937 ft., as recorded in column 12, and adding the known elevation at the previous station, we have the interpolated elevation of 804.057, column 13. To this is now added the superimposed curve resistance which was found equal to 0.292 in column 11, and we have the equivalent elevation of the top of rail if the track had been straight as recorded in column 14, 804.349.

37 Line 3 shows an actual elevation at station 934 of 806.12. The equivalent curve resistance 0.292 already calculated and recorded in column 11 should be added to this, making the equivalent elevation 806.412, column 15. Line 4 shows elevation 807.73 at Station 1434, and by the same process as above, column 15 gives 808.022. Line 5 records the end of the tangent at Station 1480 which is 46 ft. beyond the previous point at which an elevation is available, so that the interpolated rise of +0.212, column 12, as previously explained, must be added to 807.73, giving 807.942 as shown in column 13, and this increased by the equivalent grade rise of +0.292 or an equivalent elevation of 808.234, column 14. Line 6 shows that at Station 1534 an elevation of 808.19 was recorded, which being 100 ft. in advance of the last known elevation, with a difference between the two of +0.46 ft., we have a grade of +0.46 per cent, column 6. Fifty-four feet of a 1-deg. curve gives an equivalent grade rise of +0.027 ft., and this added to the previous +0.292 makes the cumulative grade rise of +0.319 shown in column 11.

38 The succeeding steps are plain and have been previously covered in detail, the only care necessary being the proper observance of the minus signs in lines 44, 45, and 46, column 12. From the results obtained in columns 14 and

15, an equivalent profile may be plotted, and this is shown in the dotted line above the solid-line true profile in Fig. 1. This represents what the grades on a straight track would be in order that a westbound train would meet the same resistance that it does on the actual existing track with its curves. Therefore, while the difference in the elevation above sea level between QY and FR is 86.77 ft., the curves have added an apparent resistance equal to 21.467 ft. more, or a total equivalent rise of 108.237 ft. This would be a total equivalent grade of rise divided by distance or 108.237 ÷ 24,425 = +0.443 per cent. From a distance of 22,634 ft. from QY to the end of the section, the equivalent profile shows practically a level, so that the equivalent grade would be 911.160 − 803.12 = 108.04 ÷ 22,634 = +0.478 per cent. From Station 1480 with an equivalent elevation of 808.234 ft. to Station 14,953 at equivalent elevation 877.061 there is an equivalent rise of 68.827 ft. in a distance of 13,473 ft. or an equivalent grade of +0.51 per

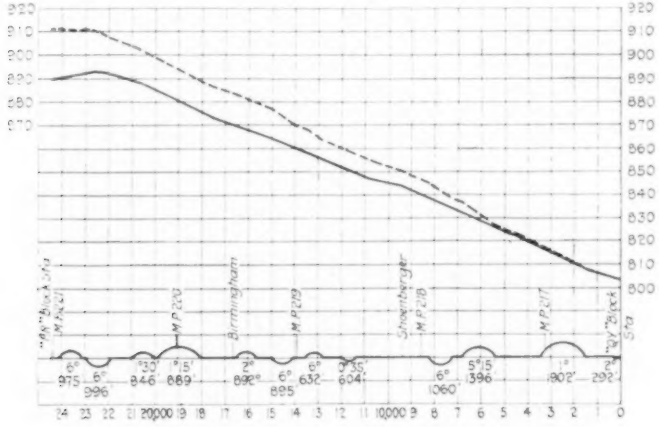


FIG. 1 PORTION OF A RAILROAD BETWEEN BLOCK STATIONS QY AND FR

cent. This distance is more than two and one-half miles, and is therefore too great to be treated as a velocity grade. From Station 5374, elevation 827.270, to Station 8230, elevation 845.734, a distance of 2856 ft., there is an equivalent rise of 18.464 ft. or an equivalent grade of +0.648 per cent. The distance is only a little over half a mile, however, or considerably less than a train length, and can therefore be operated as a momentum grade. A similar but still shorter and steeper grade occurs between Stations 7170 and 8230, where there is a rise of 7.208 ft., equivalent to a +0.68 per cent grade. This, however, is so short as to be purely a momentum grade.

39 Having followed the method above described in detail, it is proper to state that it may not be necessary to adopt this operation for an entire division. On portions of the railroad where easy grades and curves prevail, a lump-sum method may be pursued which will take the entire territory from one block station to another as a unit. In Table 1 the entire distance from QY to FR is

TABLE 1 DATA ON PORTION OF RAILROAD SHOWN IN FIG. 1

	(1) Locations: Block Stations, Mile Posts, Passenger Stations, [Survey]	(2) Total Distance, Feet [Survey]	(3) Elevation above Sea Level, Feet [Survey]	(4) Length of Grade, Feet [Sub. col. (2)]	(5) Rise or Fall in Feet [Sub. Col. (3)]	(6) Per Cent of Grade [(5) ÷ (4)]	(7) Length of Curve, Feet [Survey]	(8) Degree of Curve [Survey]	(9) Feet × Degrees [(7) × (8)]	(10) Equivalent Grade Rise, Feet [(9) × 0.05 ÷ 100]	(11) Accumulative Rise % Curve [Addn. (10)]	(12) Interpolated Rise, Feet [(6) × (7)]	(13) Interpolated Elevation [(3) + (12)]	(14) Equivalent Elevation [(11) + (13)]	(15) Equivalent Elevation [(3) + (11)]
1	"QY" Block Station.....	0	803.12	.....	0	.....	.....	2° −00′	.....	.....	.....	.....	.....	803.12	.....
2		292	.....	.....	.....	.....	292	2° −00′	584	+0.292	+0.292	+0.937	804.057	804.349	.....
3		934	806.12	934	+3.00	+0.321	642	Tangent	.....	.....	+0.292	.....	.....	.....	806.412
4		1434	807.73	500	+1.61	+0.322	.....	Tangent	.....	.....	+0.292	.....	.....	.....	808.022
5		1480	.....	.....	.....	.....	46	Tangent	.....	.....	+0.292	+0.212	807.942	808.234	.....
6		1534	808.19	100	+0.46	+0.46	54	1° −00′	54	+0.027	+0.319	.....	.....	.....	808.509
7	M. P. 217.....	3227	.....	.....	.....	.....	1693	1° −00′	1693	+0.847	+1.166	+8.296	816.486	817.652	.....
8		3382	.....	.....	.....	.....	155	1° −00′	155	+0.078	+1.244	+0.760	817.245	818.489	.....
9		4134	820.92	2600	+12.73	+0.49	.....	Tangent	.....	.....	+1.244	.....	.....	.....	822.164
10		4434	822.34	300	+1.42	+0.47	.....	Tangent	.....	.....	+1.244	.....	.....	.....	823.584
11		4734	823.20	300	+0.86	+0.29	.....	Tangent	.....	.....	+1.244	.....	.....	.....	824.444
12		4934	824.09	200	+0.89	+0.45	.....	Tangent	.....	.....	+1.244	.....	.....	.....	825.334
13		5374	.....	.....	.....	.....	400	Tangent	.....	.....	+1.244	+1.936	826.026	827.270	.....
14		6634	831.58	1700	+7.49	+0.44	1260	5° −15′	6615	+3.308	+4.552	.....	.....	.....	836.132
15		6770	.....	.....	.....	.....	136	5° −15′	714	+0.357	+4.909	+0.517	832.097	837.006	.....
16		7170	.....	.....	.....	.....	400	Tangent	.....	.....	+4.909	+1.520	833.617	838.526	.....
17		8230	.....	.....	.....	.....	1060	6° −00′	6360	+3.180	+8.089	+4.028	837.645	845.734	.....
18	M. P. 218.....	8577	.....	.....	.....	.....	347	Tangent	.....	.....	+8.089	+1.319	838.964	847.053	.....
19	"S" Pass. Station.....	9349	.....	.....	.....	.....	772	Tangent	.....	.....	+8.089	+2.934	841.898	849.987	.....
20		9434	842.22	2800	+10.64	+0.38	.....	Tangent	.....	.....	+8.089	.....	.....	.....	850.309
21		10734	846.90	1300	+4.68	+0.36	.....	Tangent	.....	.....	+8.089	.....	.....	.....	854.989
22		11350	.....	.....	.....	.....	616	Tangent	.....	.....	+8.089	+2.587	849.487	857.576	.....
23		11954	.....	.....	.....	.....	604	0° −35′	352	+0.176	+8.265	+2.537	852.024	860.289	.....

24		12850	.....	.....	.....	.....	896	Tangent	.....	.....	.....	+3.763	855.767	864.052	.....
25		13482	.....	.....	.....	.....	632	6° −00′	3792	+1.896	+10.161	+2.654	858.441	868.602	.....
26		13634	859.07	2900	+12.17	+0.42	.....	Tangent	.....	.....	+10.161	.....	.....	.....	869.231
27	M. P. 219.....	13938	.....	.....	.....	.....	304	Tangent	.....	.....	+10.161	+1.186	860.256	870.417	.....
28		14058	.....	.....	.....	.....	120	Tangent	.....	.....	+10.161	+0.4680	860.724	870.885	.....
29		14953	.....	.....	.....	.....	895	6° −00′	5370	+2.685	+12.846	+3.491	864.215	877.061	.....
30		15034	864.52	1400	+5.45	+0.39	.....	Tangent	.....	.....	+12.846	.....	.....	.....	877.366
31		15653	.....	.....	.....	.....	619	Tangent	.....	.....	+12.846	+2.450	866.970	879.816	.....
32		16545	.....	.....	.....	.....	892	2° −00′	1784	+ .892	+13.738	+3.122	870.092	883.830	.....
33	"B" Pass. Station.....	16628	.....	.....	.....	.....	83	Tangent	.....	.....	+13.738	+0.291	870.383	884.121	.....
34		17434	872.92	2400	+8.40	+0.35	.....	Tangent	.....	.....	+13.738	.....	.....	.....	886.658
35		18039	.....	.....	.....	.....	605	Tangent	.....	.....	+13.738	+2.783	875.703	889.441	.....
36	M. P. 220.....	19096	.....	.....	.....	.....	1057	1° −15′	1321	+0.660	+14.398	+4.862	880.565	894.963	.....
37		19928	.....	.....	.....	.....	832	1° −15′	1040	+0.520	+14.918	+3.827	884.392	899.310	.....
38		20056	.....	.....	.....	.....	128	Tangent	.....	.....	+14.918	+0.589	884.981	899.909	.....
39		20734	888.10	3300	+15.18	+0.46	678	1° −30′	1017	+0.508	+15.426	.....	.....	.....	903.526
40		20902	.....	.....	.....	.....	168	1° −30′	252	+0.126	+15.552	+0.571	888.671	904.223	.....
41		21912	.....	.....	.....	.....	1010	Tangent	.....	.....	+15.552	+3.434	892.105	907.657	.....
42		22234	893.20	1500	+5.10	+0.34	322	6° −00′	1932	+0.966	+16.518	.....	.....	.....	909.718
43		22634	893.44	400	+0.24	+0.06	400	6° −00′	2400	+1.200	+17.718	.....	.....	.....	911.158
44		22908	.....	.....	.....	.....	274	6° −00′	1644	+0.822	+18.540	−0.630	892.810	911.350	.....
45		23156	.....	.....	.....	.....	248	Tangent	.....	.....	+18.540	−0.570	892.240	910.780	.....
46		24131	.....	.....	.....	.....	975	6° −00′	5850	+2.925	+21.465	−2.243	889.997	911.462	.....
47		24134	890.02	1500	−3.42	−0.23	.....	Tangent	.....	.....	+21.465	.....	.....	.....	911.485
48		24334	889.89	200	−0.13	−0.07	.....	Tangent	.....	.....	+21.465	.....	.....	.....	911.355
49	M. P. 221.....	24387	.....	.....	.....	.....	.....	Tangent	.....	.....	+21.465	0.00	889.89	911.355	.....
50	"FR" Block Station.....	24425	899.89	91	0.00	0.00	38	Tangent	.....	.....	+21.465	.....	.....	.....	911.355
51		.....	86.77	.....	86.77	.....	.....	.....	42930	.....	.....	.....	.....	.....	.....

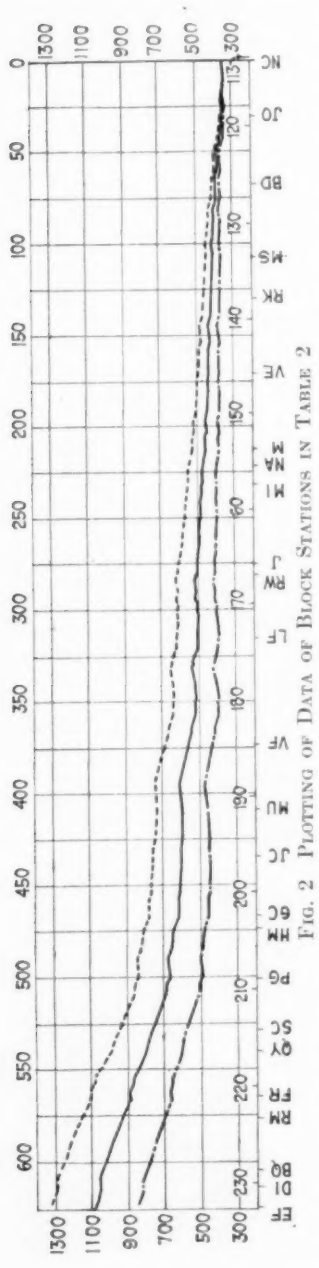


Fig. 2 Plotting of Data of Block Stations in Table 2

24,425 ft. and the difference in elevation is 86.77 ft. The sum of all the lengths of curves multiplied by the degrees is 42,930, which is the equivalent length of a 1-deg. curve. Translating this into rise by multiplying by 0.05 and dividing by 100, we have 21.465 ft.

40 Table 2 gives particulars of the block stations from the mouth of the yard at one end of the division to the entrance of the yard at the other end of the division, comprising the distances between block stations, the cumulative distance from the starting point, the actual elevations of the top of rail at the towers, the equivalent lengths of 1-deg. curves, the equivalent rise due to these curves, the cumulative equivalent rise due to curves, and the superimposed cumulative totals which give the equivalent elevations. These data have been plotted on tenths cross-section paper and are reproduced in Fig. 2, where the solid line is the actual profile, the dotted line above it the equivalent westbound grade and the dash-and-dot line below it the equivalent eastbound grade. This is a convenient profile to work to, and by making the vertical scale 100 ft. to the inch and the horizontal scale 10,000 ft. to the inch on the tenths cross-section paper, the equivalent grades may be read directly in tenths of per cent, by observing the number of tenths of an inch the grade line rises or falls in an inch. This will appear as an advantageous feature later on.

41 It will be apparent that as the curves increase the resistances encountered by a westward train on the profile shown in Fig. 2 so that the equivalent profile appears above and with steeper grades than the true profile, so also the resistance encountered by an eastward train will be greater and consequently the down grades will appear less steep for the equivalent eastward profile, and this will lie below the true profile by the same amount as the westward equivalent profile lies above it. It will also be seen that while the actual rise is from elevation 347.89 to 1088.50 or 740.61 ft., the equivalent rise due to curves is 235.389 ft. more, or an apparent rise of 1323.889 ft. On the other hand, the resistance of the curves makes the descent apparently less by 235.389 ft., or what is the same thing, we can plot the eastward profile from

a starting point 235.389 ft. below the west end of the true profile or 1088.50 – 235.389 equals 853.111. The equivalent drop for eastward trains is, therefore, only 853.111 – 347.89 = 505.221, the last half of the run being practically a level. There is an equivalent drop ofless than 40 ft. in a distance of 312,000 ft., although intermediate adverse grades of 0.1 to 0.2 per cent are encountered.

TABLE 2 DATA ON BLOCK STATIONS OF RAILROAD DIVISION

Block Stations (1)	Distance Between Block Stations (2)	Cumulative Distance from NC (3)	Actual Elevations (4)	Equivalent Length of 1° Curve (5)	Equivalent Rise due to Curves (6)	Cumulative Equivalent Rise, Feet (7)	Equivalent Elevation Westward (8)	Equivalent Elevation Eastward (9)
NC			347.89				347.89	347.89
JC	30,328	30,328	351.98	19,434	9.717	9.717	361.697	342.263
BD	36,900	67,228	388.14	17,370	8.685	18.402	406.542	369.738
MS	39,618	106,846	404.26	45,740	22.870	41.272	445.532	362.988
RK	22,847	129,693	410.90	7,394	3.697	44.964	455.864	365.936
VE	41,243	170,936	425.41	23,836	11.918	56.887	482.297	368.523
M	41,414	212,350	442.56	32,377	16.188	73.075	515.635	369.485
NA	9,579	221,929	459.39	12,437	6.218	79.293	538.683	380.097
MI	9,200	231,129	464.45	4,891	2.445	81.738	546.188	382.712
J	44,184	275,313	498.29	34,166	17.083	98.821	597.111	399.469
RW	5,407	280,720	500.16	7,068	3.534	102.355	602.515	397.805
LF	35,359	316,079	502.88	20,659	10.329	112.684	615.564	390.196
VF	58,091	374,170	559.48	34,485	17.243	129.927	689.407	429.553
MU	35,128	409,298	594.59	17,606	8.803	138.730	733.320	455.860
JC	25,524	434,822	603.04	29,324	14.662	153.392	756.432	449.648
GC	32,420	467,242	622.12	19,111	9.556	162.948	785.068	459.172
HM	7,233	474,475	648.39	3,931	1.966	164.914	813.304	483.476
PG	26,631	501,106	679.23	26,870	13.435	178.349	857.579	500.881
SC	27,954	529,060	770.33	11,304	5.652	184.001	954.331	586.329
QY	11,396	540,456	803.12	20,331	10.166	194.167	997.287	608.935
FR	24,425	564,881	889.89	42,930	21.465	215.632	1105.522	674.258
RM	11,874	576,755	935.25	20,815	10.408	226.040	1161.290	709.210
BQ	27,499	604,254	1044.61	18,090	9.045	235.085	1279.695	809.525
DI	9,001	613,255	1056.38	0	0.	235.085	1291.465	821.295
EF	11,129	624,384	1088.50	607	0.304	235.389	1323.889	853.111
Total	624,384			470,776				

APPENDIX NO. 2

LOCOMOTIVE TRACTIVE POWER AND TRAIN RESISTANCE

42 The two fundamental equations of locomotive tractive power and train resistance are:

$$T = \frac{2PC}{1 + \frac{110\,wCV}{3KH}} - \{22 + 0.15(n-1)V\}Q - 0.1V^2$$

$$R = 100N + 1.5W + 0.01(V + 16)V\sqrt{WN} + (c + 20G)W$$

43 T is in pounds, refers only to the locomotive, and is made up of: cylinder tractive force, locomotive machinery friction, and head-on wind resistance, which are respectively represented by the three terms of the right-hand member of the equation.

44 R refers to total train resistance, being exclusive of locomotive machinery friction and head-on wind resistance, but includes the remainder of the resistance of the locomotive (its rolling resistance), the total resistance of the tender and the cars behind the tender. This is done by making W the total weight of train in tons inclusive of locomotive and tender, and by making N represent a number greater than the number of cars behind tender by a constant (3) as the equivalent of the locomotive and tender in cars.

45 The other quantities, using a standard Mikado type locomotive, are:

P = nominal boiler pressure (in lb. per sq. in.) less ten = 205 − 10 = 195

C = square of the cylinder diameter in inches times the stroke in inches, divided by the diameter of the drivers in inches = $\frac{d^2s}{D} = \frac{27 \times 27 \times 30}{62}$ = 352.74 for Lls

H = heating surface, sq. ft.
(For Lls = 3731 for flues + 1147 for superheater pipes + 315 for fire box = 5193 sq. ft.)

K = evaporation (actual) per sq. ft. of heating surface per hour, assumed at 10 lb.

w = weight of 1 cu. ft. of steam, taken at 0.39 lb. corresponding to 100 degrees of superheat

c = curvature of track in degrees at 1 lb. per ton per degree of curve

G = grade in per cent at 20 lb. per ton of total W per 1 per cent

V = speed in miles per hour

Q = weight on drivers in tons of 2000 lb. (For Lls = 120.1)

n = number of pairs of drivers (For Lls = 4)

N = number of cars behind tender plus three (3), as the equivalent of the locomotive and tender

W = weight of entire train in tons of 2000 lb., including locomotive and, tender (Lls = 250 tons)

46 For 15 miles per hour the tractive power will be:

$$T = \frac{2\times195\times352.74}{1+\dfrac{110\times0.39\times352.74\times15}{3\times10\times5193}} - \{22+0.15(4-1)\times15\}\ 120.1-0.1\times15^2$$

$$= \frac{137568}{1+\dfrac{226988}{155790}} - \{22+0.15\times3\times15\}\ 120.1-0.1\times225$$

$$= \frac{137568}{1+1.47} - \{22+6.75\}120.1-22.5$$

$$= \frac{137568}{2.47} - (28.75\times120.1) - 22.5$$

$$= 55750 - 3452.875 - 22.5$$

$$= 52275 \text{ at } 15 \text{ miles per hour}$$

47 As the evaporation of 10 lb. of water per sq. ft. of heating surface per hour requires the firing of more than 8000 lb. of coal, or 8 shovelfuls per minute, and involves clean heating surfaces, good coal, clean grates, tight piston packing, etc., it is hardly fair to expect the average run of freight locomotives to be in such perfect condition. Consequently, it is proper to discount the theoretical tractive power by 10 per cent and therefore 90 per cent of 52,275 would be 47,047. It is also unreasonable to expect a locomotive to develop this high tractive power with a wet rail or in foggy or frosty weather, the coefficient of adhesion being only 5 to 1.

48 The second of the fundamental equations deals with train resistance. Using a train of 100 standard hopper cars weighing empty 25 tons each, the weight of the train back of the tender would be 2500 tons. Adding the weight of the engine and tender at 250 tons gives $W = 2750$ tons. Bearing in mind that N is three more than the number of cars, N is 103 and V, the speed, is 15 miles per hour:

$$R = 100\times103 + 1.5\times2750 + 0.01(15+16)15\sqrt{2750\times103} + (c+20G)2750$$

49 As we have already developed in Appendix No. 1, the equivalent straight track is represented by the dotted profile n Fig. 1 and therefore c becomes zero, there being no curves. G in per cent may be found by taking the equivalent elevation 808.509 at Station 1534 and 845.734 at 8230, making a rise of 37.225 ft. in 6696 ft. or +0.557 per cent. This at 20 lb. per 1 per cent would be 11.14 lb. per ton, or for the 2750 tons, 30635 lb.

$$R = 10{,}300 + 4{,}125 + 0.01(31)15\sqrt{283250} + (0+20\times0.557)2750$$
$$= 14{,}425 + (4.65\times532.3) + (11.14\times2750)$$
$$= 14{,}425 + 2475 + 30635$$
$$= 47{,}535 \text{ at } 15 \text{ miles per hour}$$

50 The above result for R is slightly larger than the corresponding value for T, but the difference is met by a slight drop in the speed of the train. Nothing remains, however, for the additional resistance due to side winds, or the inability of the engine to develop the full tractive power due to bad rail, poor coal, etc.

APPENDIX NO. 3

ADJUSTED TONNAGE RATING

51 Various methods of loading locomotives have been followed in the past, starting in the early days with a given number of cars. As the cars varied greatly in weight when loaded, and also varied in size and weight between

TABLE 3 WESTBOUND SCHEDULE BETWEEN BLOCK STATIONS NC AND DI

(1) Block Station	(2) Miles between towers	(3) Speed, miles per hour	(4) Minutes to run block	(5) Minutes standing	(6) Slow down — accelerate	(7) Total time in block	(8) Cumulative time, minutes	(9) Cumulative time, hours
NC								
	5.7	24½	14	.....	.....	14	14	0:14
JO								
	7.0	23½	18	.....	.....	18	32	0:32
BD								
	7.5	22	20½	10½	10 (water)	41	73	1:13
MS								
	4.3	26	10	.....	.....	10	83	1:23
RK								
	7.9	25	19	.....	.....	19	102	1:42
VE								
	7.8	24½	19	.....	.....	19	121	2:01
M								
	3.6	24	9	41	(fire 10 cleaned, coal and water)	60	181	3:01
MI								
	8.3	22½	22	.....	.....	22	203	3:23
J								
	1.1	22	3	.....	.....	3	206	3:26
RW								
	6.6	25	16	.....	.....	16	222	3:42
LF								
	11.1	24	27½	13½	12 (water)	53	275	4:35
VF								
	6.7	20	20	.....	.....	20	295	4:53
MU								
	4.8	24	12	.....	.....	12	307	5:07
JC								
	6.1	23	16	.....	.....	16	323	5:23
GC								
	1.4	21	4	.....	.....	4	327	5:27
HM								
	5.0	22	13½	16½	(ashpan 10 cleaned, water)	40	367	6:07
PG								
	5.3	16	20	.....	.....	20	387	6:27
SC								
	2.2	15½	8½	.....	.....	8½	395½	6:35
QY								
	4.6	15	18½	.....	.....	18½	414	6:54
FR								
	2.2	14½	9	.....	.....	9	423	7:03
RM								
	5.2	14½	21½	6½	7 (water)	35	458	7:38
BQ								
	1.7	14½	7	.....	.....	7	465	7:45
DI								

themselves when empty, it was soon found that a given number of cars would sometimes result in overloaded engines and in other cases light trains which could have hauled more cars. This resulted in uneconomical operation, and a method is now available which insures each train made up having the same

resistance. What is known as the car factor is added to the actual weight in tons of 2000 lb. of each car in the train, this imaginary weight being added for the purpose of equating the drawbar pull of all cars, whether loaded or empty, and of course involving an increased adjusted tonnage rating for the locomotive.

52 In working out a case for any given division the use of a dynamometer car is a great convenience. However, in its absence, the equivalent profiles described in Appendix No. 1 may be used. The object of the investigation is to discover the drawbar pull of a long empty train which is just equal to the drawbar pull of a short loaded train, and from the known weight of these trains and the number of cars in each, the car factor may be determined in the formula: Car Factor = $(W_1 - W_2)/(N_2 - N_1)$, where W_1 is the weight in tons of the train of heavy cars and W_2 the light cars, and N_1 the number of cars in the first train and N_2 the number in the second train.

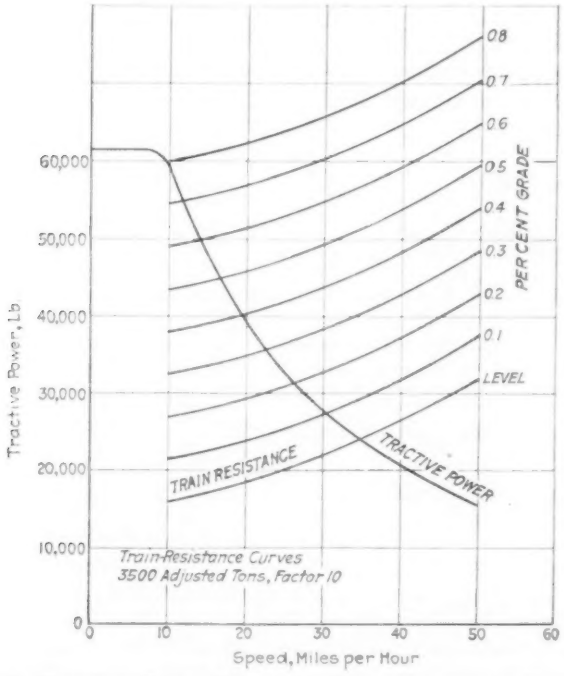


FIG. 3 TRACTIVE POWER-SPEED AND TRAIN-RESISTANCE CURVES

(Engine: Mikado, 27 × 30 in.; wheels, 62 in.; steam pressure, 205 lb.; superheat, 100 deg. fahr.; heating surface, 5193 sq. ft.; evaporation, 9 lb. per sq. ft. of heating surface per hour.)

53 An examination of Fig. 2 will indicate a very severe grade in the equivalent westbound profile between mile posts 217 and 221. This is the portion of the railroad shown on a larger scale in Fig. 1, and from Table 1 we note that the equivalent elevation at Station 1534 is 808.509 and at Station 22634 it is 911.158, a difference of 103.649 ft. in 21,100 or an equivalent grade of +0.492 per cent. The distance is four miles, which is beyond the possibility of operation

as a momentum grade. Using the method of Appendix No. 2 and assuming the long empty train to consist of 100 cars weighing 25 tons each, and substituting the values in the equation, we have:

$$R = 100\times103+1.5\times2,750+0.01(15+16)15\sqrt{2,750\times103}+(c+20\times0.492)2,750$$
$$= 10,300 + 4,125 + 4.65\sqrt{283,250} + 9.84 \times 2,750$$
$$= 10,300 + 4,125 + 2,475 + 27,060$$
$$= 43,960 \text{ at a speed of 15 miles per hour.}$$

54 By a method of trial and error with various assumed weights of a 40-car train we find that 40 cars weighing 77.3 tons each will have the same resistance. Bearing in mind that $N = 40 + 3 = 43$ and $W = 40 \times 77.3 = 3092 + 250 = 3,342$,

$$R = 100 \times 43 + 1.5 \times 3,342 + 4.65\sqrt{(143,706)} + 9.84 \times 3,342$$
$$= 4,300 + 5,013 + 1,762 + 32,885$$
$$= 43,960$$

55 The make-up of the two trains having been determined, the values of W and N may be substituted in the car-factor formula previously given in this appendix, and we have:

$$\text{Car Factor} = \frac{W_1 - W_2}{N_2 - N_1} = \frac{3,342 - 2,750}{103 - 43} = 9.866$$

56 As proof that these two trains which have an equal resistance are properly made up by using the car factor of 9.87, we may note the example below:

$$100 \text{ cars} \times 25 \text{ tons} = 2,500 \text{ tons}$$

$$100 \text{ cars} \times \text{factor } 9.87 = \frac{987}{3,487} \text{ adjusted tons}$$

$$40 \text{ cars} \times 77.3 \text{ tons} = 3,092 \text{ tons}$$

$$40 \text{ cars} \times \text{factor } 9.87 = \frac{395}{3,487} \text{ adjusted tons}$$

57 For the sake of simplicity and ease of working by yard clerks and yard masters in making up trains, we will use the nearest whole figure to 9.87 and call the adjustment factor 10. This will make up the long empty train of 100 cars having a weight of 25 tons each plus the factor 10 per car or total of 3,500 adjusted tons. The short train with 77.3 ton cars with the factor added, becomes 87.3 times 40 cars or 3,492 adjusted tons. The slight discrepancy is caused by taking the nearest whole number to the factor worked out in decimals and does not produce as much of an error as one car in the loading.

58 The good-weather rating for the engine used in this discussion has therefore been fixed at 3,500 adjusted tons with a factor of 10 westbound.

59 The car factor varies with the physical characteristics of different divisions, as a general proposition the steeper the grades the lower being the figure. On some divisions the car factor is 2. On others it is as high as 80.

APPENDIX NO. 4

TRACTIVE POWER–SPEED CURVES AND ADJUSTED-TONNAGE TRAIN-RESISTANCE CURVES

60 In plotting the locomotive tractive power the formula mentioned in Par. 4 may be used up to speeds of six or eight miles per hour, namely: Tractive power = $0.85pd^2l/D$. In the engine we have been discussing, tractive power would accordingly be $0.85 \times 205 \times 27 \times 27 \times 30/62 = 61{,}465$.

61 This may be plotted on tenths cross-section paper from the axis of zero speed to, say, eight miles per hour, as shown in Fig. 3.

62 In Appendix No. 2 the value of T was calculated for 15 miles per hour and was found to be 52,275. Repeating the calculations for speeds of 20, 25, 30, 40 and 50 miles per hour, we find corresponding values of T as 43,040, 36,093, 30,820, 22,996 and 17,215. Ninety per cent of these values for everyday working would give:

Speeds	0	15	20	25	30	40	50
T	61,465	47,047	38,736	32,484	27,738	20,696	15,494

These values of T may be plotted on the respective speed ordinates and a curve drawn through the points.

63 In Appendix No. 2 the fundamental equation of R contains four terms. The first and second contain no variables depending on speeds, curves or grades. Using the 100-car empty train previously calculated, we may consolidate $100N + 1.5W$ into 14,425 and for straight level track the last term becomes zero.

64 The coefficients of the third term may be simplified and we may write —

$$R = 14{,}425 + 5.323V(V + 16) \text{ for straight, level track}$$

With V = ten miles per hour,

$$R = 14{,}425 + 5.323 \times 10(10 + 16)$$
$$= 14{,}425 + 5.323 \times 260$$
$$= 14{,}425 + 1{,}384$$
$$= 15{,}809$$

Fifteen miles per hour has already been calculated as 14,425 + 2,475 or R = 16,900.

65 Repeating the calculations for speeds of 20, 25, 30, 40 and 50 miles per hour, we find corresponding values of R as 18,255, 19,885, 21,775, 26,355 and 31,985. These speeds and resistances in tabular form are as follows:

Speeds	10	15	20	25	30	40	50
R	15,809	16,900	18,255	19,885	21,775	26,355	31,985

66 These values of R may be plotted on the respective speed ordinates and a curve drawn through the points. This curve will be the resistance of a 3500-adjusted-ton train on straight level track. It will be noticed that the curve crosses the tractive-power curve at 35 miles per hour and consequently the maximum speed it is possible to obtain on straight level track with the Lls engine and a 3500-adjusted-ton train is 35 miles per hour.

67 Returning now to the fundamental equation of train resistance in Appendix No. 2, it will be noted that the fourth term contains variables due to grades and curves. By using the equivalent profiles the curve function becomes zero, and for the train under consideration made up of 100 twenty-five ton cars, $W = 2{,}500 + 250 = 2{,}750$. $20G \times W$ becomes $55{,}000G$, where G is the equivalent grade in per cent. On a 0.1 per cent grade this last term of the formula equals 5,500, which may be added to the values of R already found for straight level track at the various speeds, and these points plotted as a curve of the resistance of a 3500-adjusted-ton train on straight track with an ascending grade of 0.1 per cent.

Speeds	10	15	20	25	30	40	50
R on 0.1 per cent	21,309	22,400	23,755	25,385	27,275	31,855	37,485

68 It will be noticed that this curve crosses the tractive-power curve at about 30 miles per hour, and consequently the maximum speed it is possible to obtain on straight track with a 0.1 per cent adverse grade is 30 miles per hour with this engine and train.

69 For a 0.2 per cent grade $20G \times W$ becomes 11,000, which may be added to the values of R already found for straight level track at the various speeds and this process continued for the various grade curves shown in Fig. 3. The figures are tabulated below:

Speeds	10	15	20	25	30	40	50
R on 0.0 per cent	15,809	16,900	18,255	19,885	21,775	26,355	31,985
R on 0.1 per cent	21,309	22,400	23,755	25,385	27,275	31,855	37,485
R on 0.2 per cent	26,809	27,900	29,255	30,885	32,775	37,355	42,985
R on 0.3 per cent	32,309	33,400	34,755	36,385	38,275	42,855	48,485
R on 0.4 per cent	37,809	38,900	40,255	41,885	43,775	48,355	53,985
R on 0.5 per cent	43,309	44,400	45,755	47,385	49,275	53,855	59,485
R on 0.6 per cent	48,809	49,900	51,255	52,885	54,775	59,355	64,985
R on 0.7 per cent	54,309	55,400	56,755	58,385	60,275	64,855	70,485
R on 0.8 per cent	59,809	60,900	62,255	63,885	65,775	70,355	75,985
R on 0.9 per cent	65,309	66,400	67,755	69,385	71,275	75,855	81,485

APPENDIX NO. 5

FREIGHT-TRAIN TIME TABLES

70 In laying out a slow freight-train schedule, the equivalent profile shown in Fig. 2 is used, constructed on tenths cross-section paper, the vertical scale being 100 ft. to the inch, and the horizontal scale 10,000 ft. to the inch. Any grade may be read by using a pair of triangles to carry the slope to an intersection of the inch lines and the grade in tenths of per cent determined by counting the tenths inch spaces on the opposite vertical side of the inch square, and fractions of the tenths may be interpolated by the eye. For greater accuracy, the line may be carried along ten inches horizontally and the number of inch spaces vertically from the horizontal line through the intersection will give the tenths per cent, the tenths of an inch will give the hundredths per cent and the thousandths may be interpolated with the eye.

71 By taking in succession the various portions of the track from the starting point to the end of the run, the grades observed on the equivalent profile may be considered in connection with the tractive power-speed curves shown

in Fig. 3 and an estimate formed of the speed at which the given portion under consideration should be run, bearing in mind the condition of the fire, which will be dirtier the farther along we get from the start of the run, or the distance the coal pile is back from the fire door if coal pushers or stokers are not used, the physical condition of the fireman, who becomes more tired near the end of the trip, etc.

TABLE 4 SCHEDULE FOR 10,000 ADJUSTED TONS, FACTOR 30, EASTBOUND TRAINS

Block Station	Miles between towers	Speed, miles per hour	Minutes to run block	Minutes standing	Slow down — accelerate	Total time in block	Cumulative time, minutes	Cumulative time, hours
EF								
	2.1	18	7	.....	.....	7	7	0:07
DI								
	1.7	18½	5½	.....	.....	5½	12½	0:12½
BQ								
	5.2	19	16½	.....	.....	16½	29	0:29
RM								
	2.2	20	6½	.....	.....	6½	35½	0:35½
FR								
	4.6	20	14	.....	.....	14	49½	0:49½
QY								
	2.2	20	6½	.....	.....	6½	56	0:56
SC								
	5.3	21	15	.....	.....	15	71	1:11
PG								
	5.0	22	13½	16½	10 (water)	40	111	1:51
HM								
	1.4	21	4	.....	.....	4	115	1:55
GC								
	6.1	23	16	.....	.....	16	131	2:11
JC								
	4.8	22	13	.....	.....	13	144	2:24
MU								
	6.7	20	20	.....	.....	20	164	2:44
VF								
	11.1	18	37	12	8 (water)	57	221	3:41
LF								
	6.6	22	18	.....	.....	18	239	3:59
RW								
	1.1	22	3	.....	.....	3	242	4:02
J								
	8.3	21½	23	.....	.....	23	265	4:25
MI								
	3.6	22½	9½	47	(fire 10½ cleaned, coal and water)	67	332	5:32
M								
	7.8	21	22	.....	.....	22	354	5:54
VE								
	7.9	21½	22	.....	.....	22	376	6:16
RK								
	4.3	21½	12	.....	.....	12	388	6:28
MS								
	7.5	21	21½	8½	10 (water)	40	428	7:08
BD								
	7.0	21	20	.....	.....	20	448	7:28
JO								
	5.7	20	17	.....	.....	17	465	7:45
NC								

72 It may be convenient to use the sections between block stations as the units for these speed studies, and Table 3 shows a westbound schedule, column 1 giving the towers, column 2 the distance between block stations, column 3 the speed through the block, column 4 the minutes to run the block, column 5 the minutes standing still at water, coal or fire-cleaning stations

column 6 the minutes required for slowing down to stop at the above points and to accelerate the train again, column 7 the total time between block stations, column 8 the cumulative time in minutes and column 9 the cumulative time reduced to hours and minutes. In converting speed in miles per hour entered in column 3 into minutes to be entered in column 4, it is very convenient to use a slide rule and set the number of miles per hour in column 3 on the A-scale over the 60 mark on the B-scale and under the number of miles in column 2 on the A-scale read the number of minutes on the B-scale.

73 In the case we have heretofore discussed of the 3,500 adjusted tons, westbound train with the Mikado locomotive, we come out of the yard and pass NC Block Station at a moderate speed and find practically a level grade to JO over a somewhat rolling profile containing one adverse grade a mile and three-quarters long of 0.15 per cent. Up the grade the engine can make a speed of only 28 miles per hour as a reference to Fig. 3 will show on the drawbar pull curve half way between the 0.1 and 0.2 per cent grade lines. With the slow approach and this hill to interfere, we may safely say that an average speed of between 24 and 25 miles per hour should be made through this block.

74 Between JO and BD we have an overall adverse grade of 0.12 per cent with another mile and three-quarters grade of 0.28 per cent near the middle of it, and the necessity of shutting off before passing BD on account of a water stop close ahead and permitting the train to drift to a standstill. The 0.28 per cent grade would permit a speed of only 21½ miles per hour, and the slow-down would make the entire block about 23 or 24 miles per hour.

75 Between BD and MS we have an overall grade of 0.1 per cent with a water stop near the beginning of the block. There is more than a mile of 0.3 per cent grade leaving the water plug and 2½ miles of 0.2 per cent grade near the middle of the run. Allowing 10 minute for the slowing down and acceleration of the train, the block could be run at a speed of 22 miles per hour without this stop, or 20½ minutes for the distance. Adding 10½ minutes for the stop, calling the flag and getting started, makes 41 minutes total time between BD and MS.

76 By this method of analyzing the road conditions, equivalent profile, etc., we complete the schedule as shown in Table 3. From the time the train crew is ordered to report for duty, until the train clears the yard at NC, we will allow an initial terminal time of an hour and a quarter. This includes shifting out the cabin, running to the head of the yard, coupling up and making the air test, etc.

77 As there are several points where interference from passenger trains is met with at interlockings, we will allow an average of 10 minutes for these delays. The final terminal time is 45 minutes, and this includes yarding the train, picking up the cabin, running to the ashpit and releasing the crew. The total time is 9 hours 55 minutes, and this corresponds to a speed of 12½ miles per hour for the length of the run, — 124 miles. This meets the overtime limit at present in effect and permits the trains to cover the road — if they meet with no accidents or detentions — without putting the crew on overtime.

78 A similar schedule for 10,000 adjusted tons, factor 30, eastbound trains is shown in Table 4. A question may be raised as to the slow speed down the hill at the beginning of the run where the grades are steep enough to get up speeds of 50 or 60 miles per hour. It is not wise from an operating standpoint to run these heavily loaded tonnage trains at high speed when they are just coming out of a yard and with uncertain conditions concerning the lubrication of the journals. Furthermore, unduly high speeds would greatly increase the damage in case of

wrecks, and therefore the speed is restricted to a reasonable limit until the first water station is reached, and this gives the train crew an opportunity to go over the equipment and take care of any incipient hot boxes, or set off any cars that have run hot already.

APPENDIX NO. 6

DAILY SLOW FREIGHT-TRAIN PERFORMANCE RECORD

79 The record shown in Fig. 4 is prepared each morning from the train sheets and tonnage records of the previous day. Each dot or circle represents a train, its position vertically indicating on the scale at the left the percentage of the full capacity of the engine utilized, and its position across the sheet, read from the scale of hours at the bottom, shows the time the train crew was on duty. The red dots (shown in Fig. 4 as small circles) represent eastbound trains and the black dots westbound trains. The extreme left-hand dot shows a westbound train which was 99 per cent of the full adjusted-tonnage rating of the engine, and it made the run over the division in 8 hours 12 minutes from the time the crew

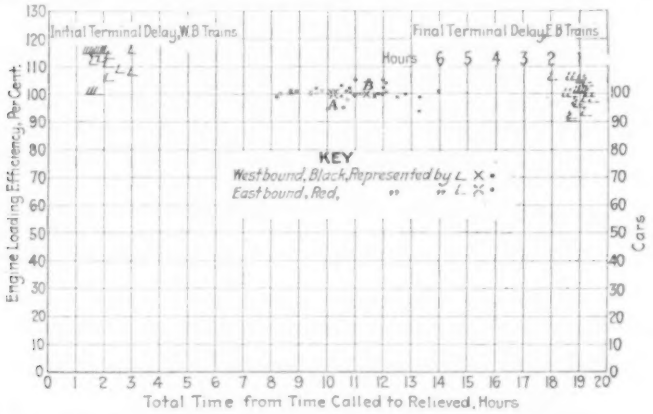


FIG. 4 DAILY SLOW FREIGHT-TRAIN PERFORMANCE RECORD

were called to report for duty until they were relieved from duty at the opposite terminal. It includes initial and final terminal delay and this particular train made an excellent run, well within the overtime limit. The red dot (small circle) slightly above and to the right represents a 100 per cent eastbound train, 8 hours and 18 minutes from called to relieved. The dot to the extreme right shows a westbound 101 per cent train, 14 hours. It is seen at a glance that the lowest loading for that day was 94 per cent and the highest 105 per cent. The spread of the time was from 8 hours and 12 minutes to 14 hours. Fourteen runs were made within the overtime limit (of ten hours and a quarter), and no crews were relieved on account of the 16-hour law. The average of all the red dots (small circles) is shown by the red cross at A and its position shows that the average of all the eastbound trains that day were loaded to 100 per cent and the average time was $10\frac{1}{2}$ hours. The black cross at B shows that the average loading of westbound trains was also 100 per cent and the time just under $11\frac{1}{2}$ hours.

80 The black characters near the left-hand margin indicate the class of engine hauling the westbound trains, the number of cars being shown on the scale at the right, and the time from called to passing out of the yard being read on the scale of hours at the bottom. From this it can be seen that there were three trains of 100 cars each, 1 hour and 24 minutes, 1 hour and 30 minutes and 1 hour and 42 minutes getting out of the yard. There were nine trains of 115 cars each, varying from 1 hour and 12 minutes to 2 hours and 48 minutes initial terminal time. Other trains are as indicated, the total number completing their runs that day being 20. Similar information is shown for the eastbound trains by the red (dotted-line) characters near the right-hand margin, the final terminal delay being recorded from right to left and read on the scale just above; the number of cars per train varying from 90 to 105, and the time from 36 minutes to 2 hours and 6 minutes. The same sheet is also used to record fast freight trains, but these have been omitted to avoid complication. An explanation can be given on the back of any unusual conditions, undue length of time on the road, reason for relieving crews, etc. A code of symbol letters reduces the need of elaborate descriptions.

81 After keeping these daily sheets for several months, the location of all the red crosses may be recorded on a sheet of tracing cloth, or a composite of all the red dots (small circles) may be made on one tracing, and through the center of gravity of all the dots a curve may be drawn, and this will show for any point on the curve the average time for the trains corresponding to that tonnage loading. A similar composite of the black dots [1] will show the westbound trains. These curves are shown in Fig. 5 and are the basis of the calculations of the most economical train loading described in Appendix No. 7. Even if most of the

[1] It is suggested that black ink be used in making the composite of the red dots and red ink for the black dots. It is much easier to see the opposite color through the thickly clustering dots of the composite, and those below the tracing cloth will not be so readily missed.

TABLE 5 MOST ECONOMICAL TRAIN LOADING

(1) Per cent engine loading W.B.	(2) Tons per train W.B. 2400 × (1)[2]	(3) Time called to relieved, hours E.B., Curve	(4) No. of trains W.B. 48000 ÷ (2)	(5) Tons per train, E.B. 140,000 ÷ (4)	(6) Per cent engine loading, E.B. (5) ÷ 7000	(7) Time called to relieved, hours, W.B., Curve	(8) Time crews on duty 1 R.T., hours (3) + (7)	(9) Total time of engine 1 R.T., hours (8) + 24	(10) Engine hours, total (9) × (4)	(11) No. of engines required (10) ÷ 24
120	2880	14.18	16.667	8400	120	12.93	27.11	51.11	851.85	35.49
115	2760	13.00	17.391	8055	115	12.11	25.11	49.11	854.07	35.59
110	2640	12.13	18.182	7700	110	11.40	23.53	47.53	864.19	36.01
105	2520	11.41	19.048	7350	105	10.78	22.19	46.19	879.83	36.66
100	2400	10.77	20.	7000	100	10.21	20.98	44.98	899.60	37.48
95	2280	10.19	21.053	6650	95	9.67	19.86	43.86	923.38	38.47
90	2160	9.67	22.222	6300	90	9.16	18.83	42.83	951.77	39.66
85	2040	9.18	23.529	5950	85	8.70	17.98	41.98	987.75	41.16
80	1920	8.70	25.	5600	80	8.26	16.96	40.96	1024.00	42.67

[2] 2400 × (1) = 2400 multiplied by value in Column (1).

dots and circles are concentrated along the lines of full tonnage, there will usually come days when engines are being loaded light to get them to the other end of the division to meet a heavy movement, and there are occasional mistakes of yard clerks which result in underloaded or overloaded trains which are not checked up until the conductor's wheel report is recorded at the end of the run.

APPENDIX NO. 7

THE MOST ECONOMICAL TRAIN LOADING

82 The curves described in Appendix No. 6 having been prepared, the data shown in Table 5 may be calculated. Column 1 gives the percentage of engine loading, from 120 per cent down to 80 per cent. The average flat tons in the eastbound and westbound trains having been recorded each day on the sheets shown in Fig. 4 and the percentage of the engine loading being also available for each day, a summary may be run up at the end of the month and a comparison drawn as to the number of flat tons per train which would correspond to a 100 per cent tonnage train. A few months of this information will settle the proper figure and this has been shown for the westbound trains at the head of column 2. This amount multiplied by the percentages in column 1 gives the tons per train in column 2. From the load-time curves in Fig. 5 may be read the average time for a westbound train loaded to 120 per cent of the engine rating, etc., and these times recorded in column 3. The summary mentioned above will also show, by extension, the total gross tons moved each day, and the average of this figure is at the top of column 4. The number of trains which it would be necessary to run to move the average day's business may be found by dividing this number of tons by the proposed tons per train in column 2 and the result

TABLE 5 — *Continued*

(12) Overtime hr. per train R.T. (8) − 20.5	(13) Punitive overtime, hr. per train R.T. (12) × 1½	(14) Total punitive overtime, hr. per day (13) × (4)	(15) Overtime, total cost per day (14) × 3.975	(16) Straight time, hours per day (4) × 20½	(17) Straight time, cost per day (16) × 3.975	(18) Total wages cost per day (15) + (17)	(19) Value of engines at 60 cts. per hour (10) × 0.60	(20) Total engine and wage cost per day (18) + (19)	(21) Per cent engine loading W.B.
6.61	9.92	165.34	657.23	341.67	1358.14	2015.36	511.11	2526.47	120
4.61	6.92	120.35	478.39	356.56	1417.17	1895.56	512.44	2408.00	115
3.03	4.55	82.73	328.85	372.73	1481.60	1810.45	518.51	2328.96	110
1.69	2.54	48.38	192.31	390.48	1552.16	1744.47	527.90	2272.37	105
.48	0.72	14.40	57.24	410.00	1629.75	1686.99	539.76	2226.75	100
.....	.....	.....	.....	431.59	1715.71	1715.71	554.03	2269.74	95
.....	.....	.....	.....	455.55	1810.81	1810.81	571.06	2381.87	90
.....	.....	.....	.....	486.34	1917.31	1917.31	592.65	2509.96	85
.....	.....	.....	.....	512.50	2037.19	2037.19	614.40	2651.59	80

recorded in column 4. The same process is followed to obtain the figure at the top of column 5 as explained for column 4 and as there will usually be the same number of eastbound as westbound trains in order to avoid running power and crews light, the tonnage per eastbound train will be found by dividing the total tons to be moved per day by the trains per day shown in column 4 and the results recorded in column 5. The total time on duty for the crews in making a round trip is shown in column 8, and from data obtained on reports described in other appendices, it has been determined that on the division under consideration, engines are off the road and in the hands of the Motive Power Department about 24 hours for every round trip they make. Consequently, 24 hours should be added to the times shown in column 8 to give the total time of an engine for a round trip as in column 9. The number of round trips made in column 4, multiplied by the number of hours per round trip in column 9 will give the total number

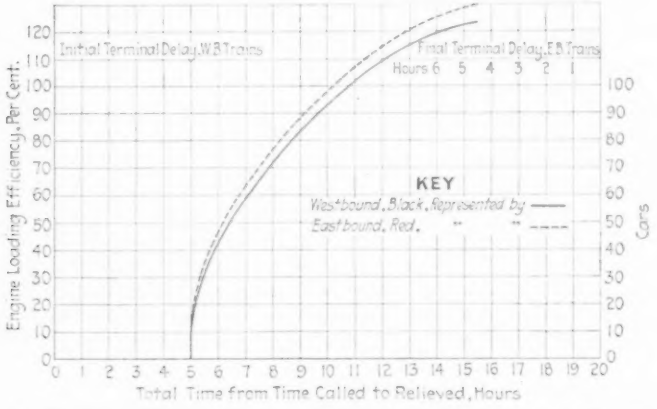


FIG. 5 TYPE OF CURVES USED TO SHOW AVERAGE TIME FOR TRAINS OF A GIVEN TONNAGE LOADING

of engine hours shown in column 10, and these figures, divided by 24 hours in the day, will give the number of engines assigned to the service as recorded in column 11.

83 As the average distance run by these trains is 128.2 miles and the overtime speed basis is now 12½ miles per hour, the time per trip is 10¼ hours and for the round trip 20½ hours. Consequently, the overtime per round trip is found by subtracting 20½ from the times shown in column 8, and the result entered in column 12. We now pay time and a half for overtime in freight-train service, therefore the number of hours shown in column 12 multiplied by 1½ gives the overtime hours for which we have to pay at the regular hourly rates, and these figures are entered in column 13. Multiplying by the number of trains per day in column 4, we have the total number of punitive overtime hours per day shown in column 14. The total wages paid to the engine and train crews in slow freight service and with the class of engine under discussion now amounts to $3.975 per hour, and the figures in column 14 multiplied by this sum gives the total overtime cost per day as shown in column 15. The straight-time hours per day

is the product of the 20½ hours for one round trip times the number of trips in column 4 and is shown in column 16. This at the rate of $3.975 per hour gives the total straight time cost per day shown in column 17, and adding the overtime cost in column 15 gives the total wages cost per day in column 18.

84 Modern Mikado locomotives of the size under consideration are worth $47,750 apiece, and taking interest at 6 per cent, depreciation at 4 per cent and insurance and taxes together at 1 per cent, we have fixed charges of $5,256 per locomotive per year, or $14.40 per day or 60 cents per hour. Column 19 shows the value of the engine hours in column 10 and the sum of the wages cost in column 18 gives the total engine and wage cost shown in column 20. It will be noted that this cost is a minimum at 100 per cent loading. The limits are rather narrow and an error of 10 per cent in overloading or underloading would cause a loss of $100 per day or $3000 per month on the amount of business handled on the division under consideration.

DISCUSSION

W. L. Bean. It happens that the matter of fixing tonnage rating is a particular hobby of the writer's and he has devoted more attention to that portion of the paper than to any other. The writer's first conclusions, however, are that the paper is a splendid exposition of practical means for overcoming losses incident to the inadequate use of equipment, and that it affords specific methods for working out improvements.

On one railroad in the past year, efforts to find what effect the condition of particular engines might have on fuel, time over the road, etc., led to some interesting results. For example, it was found, under our conditions, that a steam locomotive in express service may be performing satisfactorily to the operating officials to the extent of keeping off the delay sheet, and yet be burning 25 per cent more coal than it should, the engine being in such shape that the fireman, apparently, was the only fellow aware of the poor economy. Adjustments of the valve gear reduced coal consumption, on a thousand gross ton-mile basis from 100 lb. per thousand gross ton-miles to 80 lb. To make any program of value, there should be very careful attention given to the things which go to make a locomotive in such a shape as to operate consistently.

During a time of slack business it is easier to load engines and trains uniformly to get them over the road in good time. Time made by locomotives and trains over the road always seems to improve when there is a depression in business, and with it there is a reduction in fuel. We naturally think of the conditions in our own

neighborhood first, and therefore it is natural to refer to our own properties with which we are connected. The writer has noticed that during 1921 on the New Haven road the consumption of fuel on a unit basis in freight service was 10 per cent less than it was the previous year. Part of this has been accomplished by working out equality of loadings, which is the spirit of the author's paper. We all know the disadvantages of either underloading or overloading. There is more harm done in overloading than in underloading, because a combination of the overloaded engine with a condition of engine that is not up to par slows up the entire procession, and it is important to keep zealous division officers from going too far, just because it happens to work out all right in a few instances.

Comparison of the regulations in effect over a large number of railroads is surprising only in regard to the matter of equating tonnages. The writer found two or three years ago that a great many of the railroads were applying the equation factor which the author explained, but did not make any differentiation between the application of it on flat country and on grades, and of course it should be applied with full regard to the hills and flats.

The writer has found that the matter of temperature adjustment apparently has been overlooked, or the railroads have failed to recognize it as being of sufficient importance to justify working out differentiation as between the hills and the flats. The temperature factor, of course, applies only to the rolling friction, just as the equation factor applies to car weights.

A. J. Wood. The question of acceleration has not played an important part in the analysis which the author has made. However, there are a good many cases as in heavy passenger traffic between New York and New Haven, where the question of acceleration must be considered in train loading. The writer wishes merely to call attention to the possibilities, in such cases, of using graphical methods for the solutions.

The writer has been interested to find that by taking certain Pennsylvania Railroad trains and with the simplest data of weights and dimensions of the train, roadbed, and the profile of the roadbed, to lay out speed-time curves, and from these to predetermine the actual running time to within a few minutes of the scheduled time for any particular run.[1]

[1] One case is fully worked in Locomotive Operation and Train Control, by A. J. Wood, pp. 61 to 70.

Does the formula for tractive power given in Par. 42 show the actual superheating surface or the actual surface plus 50 per cent? The writer is under the impression that the formula has always been used heretofore with the addition of 50 per cent of the superheating surface. Taking the total heating surface (including superheater) plus 50 per cent of the superheating surface as the equivalent heating surface, will give in almost all cases a tractive-effort curve considerably higher than that obtained by other methods, and usually higher than values from the test plant results. If the results in the paper are intended to approach road conditions, it will be found that by taking the actual heating surface, the results will be found to approach average road conditions.

THE AUTHOR. Mr. Bean made a comment in regard to fuel. It was not intended to imply that fuel is not important. The point was that too much importance should not be given to small differences in fuel for loads slightly above or below what might be called a normal load. Fuel is what we used to talk of as the second differential. It is to be presupposed that the condition of the engine is satisfactory. There are fuel conservation committees on all important railroads; many railroads have officers specially delegated to watch fuel consumption and to be sure that all avoidable leaks in the consumption of coal have been overcome, and firemen properly instructed, among other things, to see that the coal does not fall off the tanks through the gangway or drop through holes in the deck-plate. Naturally, any discussion of the operation on a division must assume that these leaks are being watched and stopped to as great an extent as possible.

In connection with Mr. Wood's comment concerning acceleration, his views are undoubtedly proper and entitled to great consideration in laying out a passenger schedule, but this paper particularly applies to slow-freight-tonnage trains. The author expressly disclaimed any intention of dealing in these figures with high-class freight traffic, which, of course, also excludes passenger traffic. The acceleration which is needed to handle slow freight trains is certainly a differential, as one might say, of the third order, because in a schedule of the slow freight train laid out in one of the tables in the paper, it will be noticed that there are only four stops, one of which it is possible to eliminate by proper energy on the part of the fireman. The necessity of keeping freight trains moving is so well known that it was unnecessary to cumber the problem with the acceleration

portion of the formula for locomotive tractive power. The formula is available, and the acceleration figures are straight physical deductions and mathematical calculations.

A comment made by Mr. Wood also in regard to the usual figure which is found in locomotive descriptions, as to the heating surface being based on one and a half times the steam heating surface added to the fire-box and flue-heating surface is a matter of nomenclature. The constants in the denominator of the equation in Par. 42 take care of the actual heating surface.

No. 1816

AVOIDABLE WASTE IN LOCOMOTIVES AS AFFECTED BY THEIR DESIGN

By James Partington, New York, N. Y.
Member of the Society

The best way to overcome waste in locomotive operation is to design the locomotive in the first place so that it will fulfill the efficiency requirements of

1 A drawbar horsepower for the minimum amount of fuel
2 A drawbar horsepower for the minimum amount of weight of locomotive and tender
3 A drawbar horsepower for the minimum cost of repairs.

Fuel economy is effected by firebrick arches, superheaters, feedwater heaters, generous steam spaces, automatic stokers, etc. Minimum weight is secured by careful design of the machinery parts, use of special materials, use of the independent booster for special grades. Minimum cost of repairs is dependent on the use of as few bolted parts as possible, accessibility of parts, removability, and interchangeability. Standard replacement parts can be secured from builders.

Much waste in locomotive operation can be avoided by making a careful study of present motive-power equipment, modernizing it where necessary, or replacing by new equipment. Several examples of what has been accomplished in service by locomotives designed to yield maximum efficiency are given.

IT seems advisable to consider this subject from the constructive standpoint of indicating what constitutes good design as demonstrated by locomotives in actual service, rather than to attempt to point out the defects in locomotives which do not show maximum efficiency. If any power plant or engine is not properly proportioned for the work it has to do, the most expert skill in operation can reduce only in part the waste resulting from having such equipment in service.

2 First, considering the design of steam locomotives from the standpoint of new equipment, when a railroad company is in the market for new locomotives its requirements may be met sometimes by duplicating locomotives in service on the road, but adding

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newly developed attachments which make for increased efficiency and economy. More frequently, however, it will be found that increased traffic, change from wooden to steel cars, improvement in track, roadbed and bridges, etc., will justify and make advisable the adoption of locomotives of a larger and more powerful type.

3 Then careful consideration must be given to service requirements — maximum loads to be hauled, capacity of cars, approximate proportion of loaded to empty cars per train, grades, curves, running time over divisions, maximum allowable load per axle, location of coal chutes and water tanks, clearances, conditions under which trains must be started, and any other special requirements of the service.

4 Having determined the drawbar pull necessary, it remains to design a locomotive with the following efficiency requirements:

(*a*) A drawbar horsepower for the minimum amount of fuel
(*b*) A drawbar horsepower for the minimum amount of weight of locomotive and tender
(*c*) A drawbar horsepower for the minimum cost of repairs.

FUEL ECONOMY

5 As standard practice in modern locomotives, a sectional brick arch in the firebox and a fire-tube superheater should be applied as a means of saving fuel in any class of service. A sectional brick arch is low in first cost, easily applied and easily renewed. It usually accomplishes a fuel saving of from 10 to 12 per cent in coal-burning engines, and about 5 per cent in oil-burning engines.

6 The very general use of superheaters has gradually brought about improved conditions of cylinder lubrication, which now make it possible and desirable for the greatest economy to use a high degree of superheat, 250 to 300 deg. now being considered the best practice. A saving of 25 to 30 per cent can be obtained.

7 The use of feedwater heaters will further conserve fuel, and these are now in general use in continental Europe and are gradually being applied to locomotives in the United States. The saving that can be realized is as much as 12 per cent. The initial cost is considerable, but the effect of the feedwater heater in operation, aside from fuel economy, will be to help reduce other boiler-maintenance charges.

8 The general proportions of the boiler should also receive careful consideration. For the best results with bituminous coal,

the length of the boiler tubes should be approximately within the following limits:

SIZE OF TUBE, IN.	DISTANCE OVER TUBE SHEET
2	18 ft. 0 in. to 19 ft. 6 in.
2¼	22 ft. 6 in. to 24 ft. 6 in.
2½	28 ft. 0 in. to 30 ft. 0 in.

For many designs of locomotives, a combustion chamber can be provided, and this will help further in the economical production of steam. A generous steam space should be provided, and the throttle designed and located to secure dry steam. The evaporative capacity of the boiler should be as nearly 100 per cent of the maximum steam requirements of the cylinders as the type of locomotive will permit. Based on a 100-per cent boiler, the grate area should be sufficient to prevent the maximum coal consumption per sq. ft. of grate per hr. from exceeding —

For bituminous coal, 120 lb.
For anthracite coal (depending on size), 55 to 70 lb.

9 When the total coal consumption exceeds 6000 lb. per hr., it is generally necessary to apply an automatic stoker. These have now been so adapted to locomotive requirements that a properly designed stoker will show economy over hand firing, aside from the necessity of its use on account of the coal consumption being greater than the physical capacity of one fireman if the boiler were hand-fired.

10 The arrangement of deflector plates and netting in the smoke-box should be carefully adapted to the fuel and combustion conditions, to provide minimum fuel waste and minimum back pressure in the cylinder exhaust passages with proper provision against fire hazards which might obtain by the throwing of sparks.

11 The boiler being designed to produce steam at a minimum cost, it is now necessary to design the engine to use this steam with maximum economy.

12 The cylinder proportions and diameter of the drivers should be such as will develop maximum horsepower at the ruling speeds for train movements. The greatest horsepower of locomotive cylinders will usually be developed within a piston speed ranging from 700 to 1000 ft. per min. Therefore, if other traffic conditions will permit, the operation of trains within these limits should show the greatest operating economy.

MINIMUM WEIGHT OF MOTIVE-POWER EQUIPMENT

13 The weight on the locomotive drivers gives an engine friction, independent of other factors, of 22 lb. per ton. The desirability of avoiding excess weight on the drivers from this standpoint alone is therefore readily apparent. When the type of engine will permit, this weight should not exceed what is necessary to give a satisfactory factor of adhesion; this is usually $4\frac{1}{4}$ times the maximum tractive power. All weight in excess of this and all other excess weight and excess tender weight should be eliminated, as far as this can be done without detriment to the design of engine and tender. This applies with particular force to the machinery parts of the engine, especially those parts which affect the counterbalance. All saving in weight in these parts usually produces a similar saving in counterbalance weights and a reduction in the dynamic augment, which is very desirable from the standpoint of track and roadbed maintenance.

14 The use of special materials to keep down weight is often amply justified if repair parts can be obtained promptly when required. This, in the past, has often been the cause of delay, but it can be guarded against by carrying a few spare parts in stock ready for renewals. High-tensile alloy steel can frequently be used to advantage for driving axles, crankpins, main and side rods, piston rods, etc.

15 Occasional steep grades or hard starting conditions at stations may cut down the hauling capacity of locomotives over a division to a serious extent. In such cases the utilization of the weight on trailer trucks for additional tractive power in starting and at slow speeds may increase the capacity of the locomotive from 10 to 25 per cent, depending on the number of driving wheels and working pressure. It has been demonstrated that a separate steam engine or booster geared to the trailing axle will give this additional traction, and that it can be cut in or cut out very satisfactorily as occasion may require.

16 This is an item in economical operation worthy of consideration where hauling capacity is restricted by such limitations, and the use of an independent booster may oftentimes permit the satisfactory operation of considerably lighter locomotives for service of this character.

17 Within the limits of this paper, only the major features of

design can be outlined briefly, and only such devices as have been carefully tried out and are in successful operation are cited. The writer believes the savings mentioned are well within what may be obtained in practice.

18 Many other improvements promising further economy in the generation and use of power in the steam locomotive are contemplated and in the experimental stage, but these do not properly come under the scope of our subject as here treated.

COST OF REPAIRS

19 It has been pointed out that locomotives and tenders should be designed to produce the required drawbar horsepower with as little excess weight as possible. In this connection, however, due consideration must be given to the question of repairs.

20 The design of boilers from the standpoint of weight is practically fixed by existing boiler regulations, which provide that locomotive boilers must be operated with a factor of safety of not less than four. Practically all boilers at the present time are designed with a factor of safety of $4\frac{1}{2}$, which leaves a comfortable margin between this and the minimum allowable operating factor.

21 The maximum stresses in other parts of the locomotive must also be carefully considered, and the parts designed to keep these stresses within limits which will eliminate costly failures in service.

22 Aside from the consideration of stresses, much repair cost can be avoided by adopting designs which reduce the number of parts, as far as reasonably may be, especially where these parts must have bolted connections. Here, however, care must be taken to avoid construction which cannot readily be removed for repairs or renewals or repaired in place with reasonable facility.

23 Many roads today are giving a great deal of thought to locomotive design along these lines, having especially in mind the desirability of making the engine parts —

Accessible for oiling and inspection
Easily removable with proper shop facilities
Of the minimum number of pieces
Interchangeable with equipment now in service.

24 The repair-shop facilities must, of course, be kept abreast of the requirements; i.e., as new and larger locomotives are put in

service, turntables, cranes, machine tools, etc., must be of sufficient capacity to handle the larger equipment economically.

25 The repairs of locomotives can often be facilitated and the necessary shop equipment kept down to the minimum by securing from the locomotive builder many parts which he is able to turn out more accurately and more economically than the average railroad shop would be equipped to do. Such parts include:

Flanged sheets for boiler repairs
Flexible and ordinary staybolts
Finished bolts and nuts
Drop forgings
Packing rings for pistons and piston valves
Special equipment which requires special tools for its production.

26 Without attempting to pursue further the design of new locomotives it may be remarked that a study of the special conditions of individual railroads is necessary to secure equipment best suited to the needs of each.

OLD MOTIVE-POWER EQUIPMENT

27 Much waste in locomotive operation can be avoided by making a careful survey of present motive-power equipment which is not giving as economical or efficient service as could be obtained if the engines were modernized. This applies particularly to locomotives where the service conditions demand more power than the present equipment can economically produce. All the suggestions made in regard to the design of new equipment are applicable to a greater or less degree to old equipment, provided the old equipment is not meeting the demands of the service from a power standpoint, or is not furnishing this power economically.

28 In making a survey of this character care should be taken to determine accurately whether the old equipment will warrant the additional cost of changes and betterments necessary to convert it into up-to-date power. This can be decided by taking the number of years the engines will be retained in service and the increased net return or saving for this period as against the cost involved for changes, interest on the additional investment, increased maintenance, etc.

29 A comparison should also be made with the results that

could be realized by the purchase of new equipment best adapted for the service, as against the cost of contemplated changes in the old equipment.

30 If these comparisons show a saving in favor of modernizing the old equipment or the purchase of new equipment, every month that the engines are kept in service without doing this will result in a loss that is not recoverable. A few concrete examples of what has been accomplished in service by locomotives designed to yield maximum efficiency may be of advantage. Notable designs, for which data are available, are as follows:

Pacific type passenger locomotive No. 50,000 built by the American Locomotive Company

Decapod type freight locomotive, Class Ils, built by the Pennsylvania Railroad Company

Heavy Mallet special-service locomotive built for the Virginian Railway by the American Locomotive Company.

AMERICAN LOCOMOTIVE COMPANY ENGINE NO. 50,000

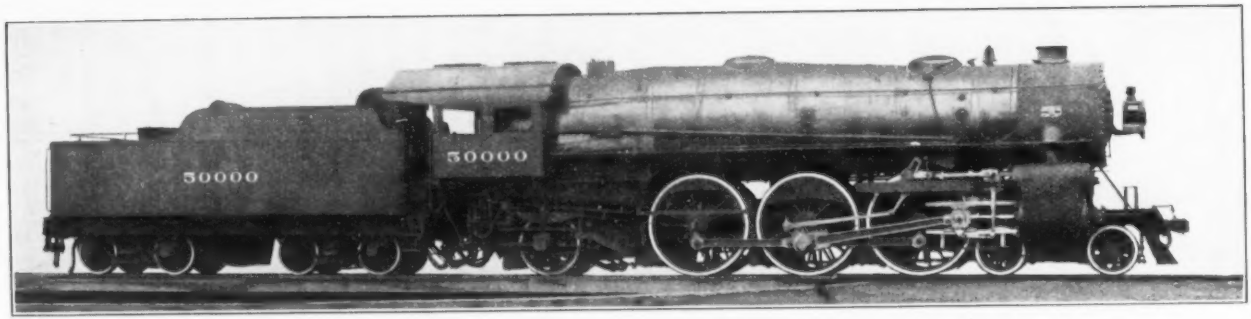
31 Locomotive No. 50,000 was built by the American Locomotive Company at the builder's expense in 1910, to demonstrate the maximum tractive power with adequate boiler capacity that could be obtained while keeping the adhesive weight below 60,000 lb. per driving axle.

32 Untrammeled by any outside specifications or the necessity of conforming to any railroad's existing standards, the builder had a free hand to embody in this design their ideas of the best engineering practice. To accomplish the purpose of the design — the maximum capacity per pound of weight — the largest boiler capacity within the predetermined wheel loads was the essential feature.

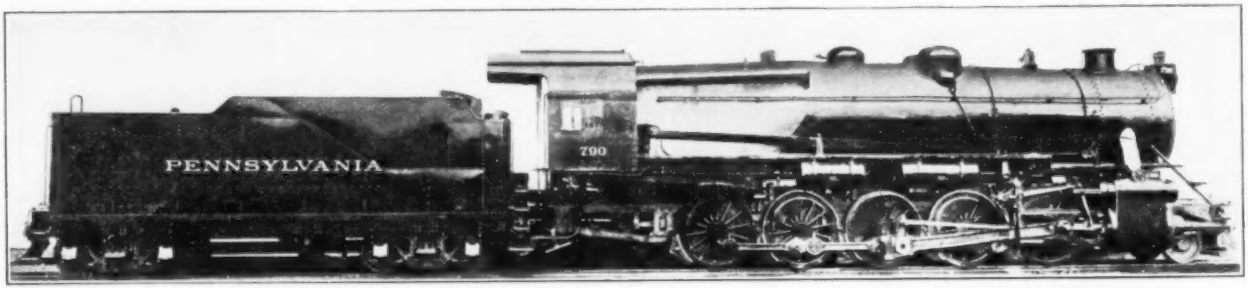
33 This end was obtained by eliminating every pound of weight in all the parts that was not necessary to strength and durability, utilizing the weight thus saved to provide a larger boiler. Then the capacity of the boiler thus secured was increased by combining in one design the most approved fuel-saving devices to obtain the utmost economy in boiler and cylinder performance.

34 Many of the large Pacific type locomotives with drivers 75 in. in diameter and over in operation today greatly exceed locomotive No. 50,000 in total weight.

35 An average of all the important engines of this type, in-



Pacific Type Passenger Locomotive, No. 50,000, Built by the American Locomotive Company



Decapod Type Freight Locomotive, Class 11s, Built by the Pennsylvania Railroad Company

TABLE 1 LARGE PACIFIC TYPE LOCOMOTIVES NOW IN OPERATION IN THE UNITED STATES

	No. 50,000												Average
Cylinder, in.	27 × 28	27 × 28	25 × 28	25 × 30	27 × 28	26 × 28	26 × 28	23½ × 26	26 × 28	25 × 28	25 × 28	25 × 28	25.6 × 28
Drivers, Diam., in.	79	80	79	75	77	79	75	79	75	77	80	75	77.5
Boiler, Diam., in.	76¾	78½	76¾	79½	75⅞	78	78⅝	70½	76	74	72	69	.....
Pressure, lb. per sq. in.	185	205	210	200	185	210	200	200	190	200	200	185	197.5
Firebox length, in.	114½	126	126	111	114⅛	126½	114½	108½	110	120½	126½	108½	.....
Firebox width, in.	75½	80	108½	84½	84	108½	84½	75½	72	84	108½	70½	.....
Tube length, ft. in.	22–0	19–0	17–3	18–6	22–0	19–0	20–6	21–6	20–0	22–0	19–0	20–0	.....
Wheelbase, driving, ft. in.	14–0	13–10	14–0	13–2	14–0	13–10	13–0	14–0	13–0	13–4	13–10	13–6	.....
Wheelbase, engine, ft. in.	35–7	36–6	36–0	34–9	36–2	35–8	34–11	36–6	34–4	35–8	35–7	34–9	.....
Weight on drivers, lb.	172500	201800	192500	181500	178000	181400	194000	184500	169500	165000	176900	168500	180500
Weight of engine, lb.	269000	308900	302000	299000	295000	291400	287000	282000	280000	279500	273600	269000	286400
Heating surface of tubes and flues, sq. ft.	3808	3746.8	2830	3232	3534.7	3454	3939	3193	3380	3720.9	2644	2970	.....
Heating surface of firebox, sq. ft.	248	288.6	351	297.6	239.8	303	259.6	231	240	266.4	282	230	.....
Heating surface, total, sq. ft.	4056	4035.4	3181	3529.6	3774.5	3757	4198.6	3424	3620	3987.3	2926	3200	3640.8
Superheating surface, sq. ft.	897	1154	645	803	962	816	970	838	830	783.5	652	778	.....
Grate area, sq. ft.	59.8	70.0	95.0	65.0	66.5	94.8	67.	56.5	55.	70.4	94.5	52.7	61.6[1] 70.6
Tractive power, lb.	40600	41845	39500	42600	41700	42770	42900	30900	40700	38600	37200	36700	39670
Cyl. hp.	2427	2690	2365	2252	2427	2556	2434	1990	2312	2252	2252	2083	2336
Boiler hp.	2235	2467	2244	2282	2104	2311	2398	1958	2112	2267	1950	1869	2186
Weight per cyl. hp, lb.	110.8	114.8	127.7	132.8	121.5	114.0	117.9	141.7	121.1	124.1	121.5	129.1	122.8
Weight per boiler hp., lb.	120.3	125.2	134.6	131.4	140.2	126.1	119.7	144.0	132.6	123.3	140.3	143.9	131.8
Boiler percentage	92.0	91.7	95.0	101.0	86.7	90.5	98.5	98.2	91.5	100.5	86.8	90.0	93.5

[1] Average of the soft-coal-burning engine only; 70.6 is the average for all the engines, including the hard-coal burners.

cluding locomotive No. 50,000, shows approximately 1000 lb. less tractive power with an increase of 17,400 lb. in weight, with the very slight advantage of only 1½ per cent in boiler capacity. (See Table 1.)

36 Locomotive No. 50,000 delivers 1 cylinder hp. for every 110.8 lb. of weight and 1 boiler hp. for every 120.3 lb. of weight. In actual tests it developed:

An average rate of 2.21 lb. of coal per i.hp-hr.
A low rate on one test of 2.12 lb. of coal per i.hp-hr.
An average rate of 16.85 lb. of steam per i.hp-hr.
A low rate on one test of 16.5 lb. of steam per i.hp-hr.
A maximum indicated horsepower of 2216 or 1 hp. for every 121.4 lb. of weight.

37 The thought occurred that possibly No. 50,000 was built too light, and that later on, in order to keep the engine in service, many of the parts might require strengthening.

38 Locomotive No. 50,000 was purchased by the Erie Railroad and numbered 2509. Mr. Wm. Schlafge, mechanical manager of the Erie, states that it has been necessary to make very few changes on the locomotive since it was received. The guide yoke was reinforced on account of working. Guide-yoke blocks were also made solid on the guide yokes. The trailer spring sliding block was changed to the same type as used on their K–4 Pacific type locomotives. No other changes or alterations have been made. Yet from the time this locomotive was placed in service on the Erie Railroad up to March 1, 1920, it had made a total mileage of 351,800.

39 Ten years of service coupled with 350,000 miles of running demonstrate the strength of the design and the figures given indicate remarkable performance.

PENNSYLVANIA RAILROAD DECAPOD TYPE FREIGHT LOCOMOTIVE, CLASS IL-S

40 While the design of engine No. 50,000 represents the best practice of the present day as measured by the economical operation of passenger locomotives, the development of heavy freight power involves the consideration of other factors materially affecting the design. In 1915 the Pennsylvania Railroad found that for the economical operation of their lines a tractive power about 25 per cent in excess of the Mikados then in use was desirable. In working on the design for such an engine, an attempt was made to

obtain better economy in performance by a radical departure in cylinder proportions. The accepted practice in proportioning cylinders is to arrange for a cut-off of nearly 90 per cent of the stroke, so that the starting torque may be as uniform as possible.

41 As the adhesive weight limits the cylinder diameter if excessive slipping is to be avoided, it is obvious that on long grades, where the maximum tractive effort is required, the long cut-offs use steam in a most uneconomical manner. The Pennsylvania Railroad having several such long grades on its lines, the new design adopted involved a limitation of the cut-off to about 50 per cent, in place of 90 per cent, and an increase in cylinder diameter to give sufficient torque at this cut-off to utilize fully the adhesive weight.

42 The expected increase in economy of coal and water due to the shorter cut-off has been fully realized. Not only has the engine shown remarkable efficiency, but the economy under wide ranges of load is especially remarkable.

43 Fortunately there is available a very complete test of this engine, made in the testing plant at Altoona (Bulletin 31, P.R.R. Testing Plant 1919). This test shows a water rate of 15.4 lb. per i.hp-hr., with a total i.hp. of 3080 at 40 per cent cut-off and a coal consumption 2.9 lb. The lowest coal consumption recorded is 2.00 lb. per i.hp-hr., obtained at an output of 1777 i.hp. and a cut-off of 30 per cent.

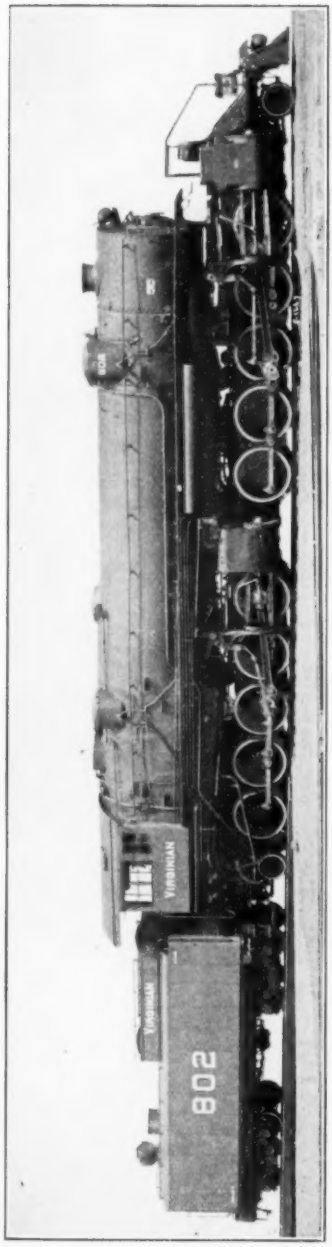
44 The thermal efficiency of the locomotive is also high and well sustained over a large range, a maximum of 8.1 being attained at an output of 1777 i.hp., and the range being from 6.1 per cent at 776 i.hp. to 5.3 per cent at 3486 i.hp. with an average of over 7 per cent for the usual operating conditions.

45 The highest drawbar pull recorded in these tests is 76,211 lb. at a speed of 7.4 m. p. h., but in road service a pull of 80,640 lb. has been recorded at 7.2 m. p. h. The indicated tractive effort plotted from a card taken at 7.4 m. p. h. at 55 per cent cut-off is slightly over 90,000 lb.

46 This design gives a calculated figure of 88.9 lb. per cylinder hp., the lowest on record. During the tests an indicated horsepower of 3486 was developed, giving a weight of 106.2 lb. per hp. The weight per boiler hp. does not compare as favorably, however, as it is 145.4 lb. The Belpaire firebox contributes materially to this excess.

TABLE 2 MODERN MAXIMUM EFFICIENCY LOCOMOTIVES

	No. 50,000		
Road	Erie	Virginian	Pennsylvania
Type	462	210102	2100
Fuel	Bituminous coal	Bituminous coal	Bituminous coal
Boiler, type	Conical connection	Extended wagon top	Conical connection
Boiler, diam., in.	76½–87	105½–118½	87–90½
Weight on drivers, lb.	172500	617000	342050
Weight on truck, lb.	49000	32000	29750
Weight on trailer, lb.	47000	35000	.....
Weight, Total, lb.	269000	684000	371800
Driving wheel diam., in.	79	56	62
Cylinders, in.	27 × 28	30 and 48 × 32	30½ × 32
Boiler pressure, lb. per sq. in.	185	215	250
Tractive power, lb.	40600	147200	90000
Factor of adhesion	4.25	4.08	3.80
Cyl. hp.	2427	5040	4182
Grate, length and width, in.	114 × 75½	144 × 108½	126 × 80
Grate, area, sq. ft.	59.7	108.7	70.01
Tubes, number	207	381	244
Tubes, length, ft. in.	22 — 0	25 — 0	19 — 1
Tubes, spacing, in.	¼	¼	½
Tubes, thickness	No. 11 B.W.G.	No. 11 B.W.G.	0.125 in.
Tubes, diam., in.	2¼	2¼	2¼
Flues, number	36	70	48
Flues, diam., in.	5½	5½	5½
Flues, thickness	$\frac{3}{16}$ in.	No. 9 B.W.G.	0.18 in.
Combustion chamber length, in.	.....	36	42
Brick arch	Security	Gaines	Security
Heat. surf. firebox, sq. ft.	248	532	290
Heat. surf. tubes, water side, sq. ft.	2672	5592	2731
Heat. surf. flues, water side, sq. ft.	1136	2511	1313
Heat surf. total, sq. ft.	4056	8635	4334
Boiler hp.	2250	4800	2553
Steam rate, calculated lb. per hp-hr.	20.8	19.7	20.8
Coal rate, calculated lb. per hp-hr.	3.25	3.1	3.25
Superheater — number of units	36	70	42
Superheater — diam., in.	1½	1½	1½
Superheater — heating surf. sq. ft	879	2120	1418
Tender weight in running order, lb.	161500	214300	182000
Tender capacity coal, tons	14	12	17½
Tender capacity, water, gal.	8000	13000	9000
Weight of locomotive, lb. per cyl. hp.	110.6	135.7	88.9
Weight of locomotive, lb. per boiler hp.	119.6	142.5	145.4
Best actual performance:			
Steam rate, lb. per hp-hr.	16.5	.....	15.4
Coal rate, lb. per hp-hr.	2.12	.....	2.0



HEAVY MALLET SPECIAL-SERVICE LOCOMOTIVE BUILT FOR THE VIRGINIAN RAILWAY BY THE AMERICAN LOCOMOTIVE COMPANY

VIRGINIAN 2–10–10–2 TYPE LOCOMOTIVES

47 The large 2–10–10–2 Mallet engines for the Virginian Railway were designed to meet the unique conditions on this road, which was built as an outlet to certain bituminous coal fields of West Virginia. Practically the entire revenue business is confined to hauling coal to the shipping docks at Sewall's Point, the westbound revenue freight being negligible in amount, as only one town of any importance, Roanoke, is located on the line.

48 As the development of the coal fields proceeded, the tonnage to be handled increased rapidly, rising from 2,141,009 in 1911 to 7,621,555 in 1920, and in order to handle the business at a profit the maximum attainable capacity in motive power was demanded.

49 Having fixed on 100 cars as the maximum number that could safely be handled in a single train, and increased the car capacity to 120 tons, it was estimated that a locomotive of 147,000 lb. tractive power would be needed to haul the train from Princeton to tidewater, a helper being used for a grade of 0.6 per cent ten miles long over the Alleghenies.

50 The 2–10–10–2 Mallets were designed to meet these conditions and their operation has been very successful. They have

handled trains of 16,000 tons on a 0.2 per cent grade with the lowest consumption of coal per ton-mile ever recorded. Unfortunately, accurate tests of coal and water per dynamometer horsepower are not available, owing to the fact that there is no dynamometer of adequate capacity to be had at present. However, on May 25, 1921, a train of 15,725 tons behind the tender was hauled from Princeton to Roanoke at a rate of 26.9 lb. of coal per 1000 ton-miles, and on May 27 a 75-car train of 12,070 tons showed the same figure for coal per thousand ton-miles.

51 One of these engines has hauled a train of 110 cars weighing 17,250 tons from Victoria to Sewall's Point, which is believed to be the heaviest train ever handled by one engine. The ruling adverse grade was 0.2 per cent.

52 The principal dimensions of the three locomotives cited and a comparison of the horsepower characteristics, calculated by the American Locomotive Company's method, are embodied in Table 2.

DISCUSSION

Elmer A. Sperry. In Germany, the great Krupp engineer Otto Alt, has written: "Among all the new and difficult problems facing Diesel engine builders, that of the Diesel locomotive takes the first place, and yet the fuel efficiency is so striking when compared with steam, that the Diesel locomotives will come to the front in spite of all technical difficulties."

The difficulties of designing a Diesel locomotive are tremendous, but we know that things which seem common-place today were considered to be almost impossible yesterday. If we can get the same drawbar pull in an 80-ton compound-Diesel locomotive that we have in the present steam locomotive, this is an encouraging outlook, because it can be done on one-tenth of the fuel being used now.

On thousands of miles of roads in this country bunker oil is being used. Formerly with the Diesel engines it was necessary to use a high-grade oil, known as Diesel oil, rather than bunker oil, which costs more on a parity with kerosene. Now we can burn bunker oil in the new compound engine.

The writer believes that the Diesel locomotive is to come; possibly it will come through electricity giving us all the advantages of electric traction, which are many.

As to fuel economy, in an 80-ton locomotive it looks as if the switching in freight yards would be done for less than three barrels of oil for each twelve hours. The cost of that oil on the New York market about the first of December 1921 was $1.35 a barrel.

H. N. Davis. Fuel economy is a thing which the writer talks about a lot, and yet he wonders whether practical people really are as much interested in fuel economy outside of these meetings as they are good enough to say they are here. For instance, the author is interested primarily in fuel economy. The first thing to look after is the locomotive which is now operating at full gear at slow speeds, and which should be put back to a 70 per cent cutoff. In some of the Pennsylvania tests, class ILS locomotives, there is shown a saving of 19 per cent only on the drawbar pull. At 7.5 miles an hour the cutoff can be pulled back from full gear to about 70 or 72 per cent cutoff on that one particular locomotive, with a saving of nearly 20 per cent on the steam consumption with a drop in drawbar pull of less than 2 per cent.

The ton-mile is worth 6 cents clear to the railroad, and multiplying so many ton-miles by six cents, and on the other side of the sheet putting on the saving of the fuel, the saving in fuel disappears and it does not pay to save fuel on that basis.

Do the railroads want maximum drawbar horsepower-hours for 100 lb. of steam or do they want maximum drawbar horsepower out of train crew or per locomotive unit?

O. C. Cromwell. We have been listening to the theoretical end of the proposition; there is something further that we can consider. The theory of design is advancing, but if we give some attention to increasing the life of some of the parts that are expensive to maintain in locomotive construction, we will go quite a considerable way toward effecting economy.

Not long ago the writer had occasion to look after the lubrication of the tenders, trailers, and the engine trucks of some locomotives, and at that time he thought he would try out some special kind of lubricant. The results of lubricating the tender and the trailer trucks were successful but when it came to engine trucks there was trouble.

The division master mechanic and the writer did not quite agree as to the cause of the troubles. The writer was surprised to find that part of the trouble was due to design. It was found that at each engine terminal the engine truck would have to be

repacked to insure a safe journey. The writer could not understand this and got down under the locomotive and placed his hands on the journals to see how hot they were; they were warm, all right. In analyzing the question, it was found that in the tender truck box there were nine pounds of packing, in the trailer trucks about sixteen pounds, and in the engine trucks about three pounds. The engine truck boxes as now designed have a heavier duty to take care of, and with a less quantity of lubrication. Although this represents standard design, one way to avoid waste is by properly lubricating the parts of the locomotive, and cutting down unnecessary and frequent attention which is required today to avoid trouble.

Another way to bring about economy is to take care of some of the repair parts and see that they do not require renewal in as short a time as at present; for instance, it is possible to increase the size of the hubs of the driving wheels and increase the bearing face of the box against the wheel, thereby insuring a longer life before the accumulation of end wear necessitates the renewing of the face of the box. Case-hardened bushings can be used in a great many places where pins are now used in plain holes, thus keeping down the cost of maintenance, by the removal of pins and bushings and saving the plugging of holes and redrilling.

We have been considering the question of fuel economy. The writer believes we can improve the stoker so that we can save fuel in that direction.

As to the question of design, if the designer could in some way follow up the performance of the design, and then improve along that line, we would get some improvement.

C. J. Mellin. The writer will only call attention to the fact that the formula from which the coal consumption per horsepower of articulated compound locomotives was arrived at was not based on the Virginian engine under consideration, but upon one of the earliest pushing engines which were designed for slow speed to suit that service and put in regular road service for test at speeds at which the contracted steam passages, etc., became inadequate and partly choked the engine with the higher piston speed in such service. This formula has not been used for any other purpose than determining the proportion of the boiler, which makes it about 15 per cent smaller than that for a simple engine of the same tractive power, although a higher percentage in fuel and water consumption is obtained.

As this has been working very satisfactorily no test has been made of more recent engines or any attempt made to produce any but for moderate speed of this type of engines.

W. F. KIESEL, JR. The paper is of particular interest, due to its presentation of locomotives designed on basic values adopted by different designers, but compared on American Locomotive Company's formulae.

Such formulae, being empirical, must be changed from time to time, to keep pace with new theories introduced in locomotive design. Even then, they are useful only as a preliminary approximation of desired values.

The author refers, in Par. 8, to 100 per cent maximum steam requirements of cylinders. On referring to American Locomotive Company's Bulletin 1017, this, for superheater locomotives, is found to be based on a horsepower $= 0.0229PA$, in which P = boiler pressure, and A = cylinder area. That formula is reasonably satisfactory, for locomotives having equal cutoff in full gear, but falls short of forming a basis on which to compare a locomotive with 90 per cent cutoff with one having 50 per cent maximum cutoff.

The test of the decapod locomotive, Class "ILS," showed that in full gear, at low speed, the steam per indicated horsepower was 38 per cent less than the steam rate for a locomotive with 90 per cent cutoff. Under average service conditions, the saving in steam is at least 15 per cent.

If the empirical formulae are changed to meet cut-off effect, such as obtains in the Pennsylvania System "ILS" locomotive, some of the values in Table 2 would be affected. With a saving of steam of 15 per cent the steam rate (calculated) in pounds per horsepower-hour, which is given as 20.8 in Table 2, would be 17.68, which compares favorably with test results. The best actual test for steam rate per indicated horsepower was 14.9 pounds.

The empirical formulae also fall short for comparison, due to relative freedom of draft, especially in the smokebox, which seriously affects the size of nozzle and cylinder back pressure.

Furthermore, the beneficial effect of large combustion chamber volume has not been clearly demonstrated. The freedom of draft and large combustion chamber volume greatly affect the maximum boiler horsepower.

The locomotives listed in the first and second columns of Table 1,

are respectively, the American Locomotive Company No. 50,000, and the Pennsylvania System Class "K4S." The test results are given for the former in Par. 36 and for the latter in Pennsylvania System Bulletin No. 29. They are, as follows:

	No. 50,000	K4S
Low rate, one test, coal lb. per i.hp.	2.12	1.52
Low rate, one test, steam lb. per i.hp.	16.5	14.96
Maximum i.hp.	2216	3184
Weight of locomotive in lb. per max. i.hp.	121.4	97.0

This shows that the "K4S" is actually far ahead of the No. 50,000, on every count, instead of being inferior, as the comparison, based on the antiquated empirical formulae, would indicate.

Although these figures show the fallacy of empirical formulae, in serving as a basis for comparative tabulation, they do not detract from the substance of the matter presented by the author, which shows the method of procedure and the strides made in design and construction during the past few years to make the locomotive a truly economical power plant.

JOHN L. NICHOLSON. The writer wishes to express his hearty appreciation of the broad and constructive basis for consideration afforded by the paper. The author has succeeded in indicating what constitutes good design as demonstrated by locomotives in actual service, and is to be thanked and commended for his selection of notable locomotives of highly developed constructions.

Among the newly developed locomotive attachments that make for increased efficiency and economy, the thermic syphon may be mentioned, by reason of its already ascertained influence over the subject of avoidable waste in locomotive design and operation.

In all tests that the railroads have made thus far, and there have been many, the thermic syphon has never failed to reduce the amount of fuel consumed per drawbar horsepower. In some cases the installation of these syphons has resulted in a 25 per cent fuel saving. An average saving of 15 to 19 per cent, according to class, is now well established.

The second efficiency requirement proposed by the author is that a drawbar horsepower be produced for the minimum amount of weight of locomotive and tender, and from the figures presented

it appears that the weight per boiler horsepower for large Pacific-type locomotives now in operation ranges from 119.6 to 144 pounds. The low weight of 119.6 lb. per boiler horsepower applies to locomotive No. 50,000 which was built by the American Locomotive Company at the builders' expense in 1910 to demonstrate the maximum tractive power that could be obtained with the largest possible boiler capacity while keeping the adhesive weight below 60,000 lb. per driving axle. The end was obtained by eliminating every pound of weight in all parts not necessary to strength and durability, and utilizing the weight thus saved to provide a large boiler. The capacity of the boiler thus secured was further increased by combining in one design the most approved fuel-saving devices then existing.

Locomotive No. 50,000 was built back in 1910; and the author therefore refers to a number of newly developed attachments which make for increased efficiency and economy. With these, the thermic syphon, also of fairly recent development, should be considered. Indeed, consideration of thermic syphons, as applied to locomotive No. 50,000, will at once demonstrate the fundamental character of the improvement which the thermic syphon has accomplished in locomotive design. The thermic syphon is an inverted triangular water leg placed vertically above the fire in the firebox. By thermal action it draws water from the throat and barrel of the boiler, and discharges it through and above the crown sheet of the firebox. Thereby the firebox heating surface is much increased and a vigorous fore-and-aft circulation of boiler water is set up. Both factors contribute to a marked increase of capacity, as well as economy. Incidentally the syphons support the brick arch. Assume that the boiler horsepower of this locomotive is 2250, as stated in Table 2, and that all the author's figures relating to boiler horsepower are based upon the formulae devised by F. J. Cole of the American Locomotive Company and that this method is used in estimating the capacity increasing ability of the thermic syphon.

Then, if two syphons were applied to locomotive No. 50,000, they would add approximately 62 sq. ft. to the radiant heat absorbing surface of the firebox. The effect of this installation would be to add 164 boiler horsepower to the capacity of the locomotive, although this is, at best, a theoretical approximation which in actual practice is far exceeded, due to the improved circulation effected throughout the entire boiler.

Allowing for the net additional weight of the syphons, the result of such installation in locomotive No. 50,000 would be to re-

duce the weight of this locomotive to 113 lb. per boiler horsepower as compared with 119.6 lb. without syphons.

Further in the case of locomotive No. 50,000, the application of thermic syphons as described would increase the boiler horsepower to 100 per cent of the cylinder horsepower. This increase in boiler capacity is alone sufficient to insure a substantial improvement in fuel economy; but assuming that a 92 per cent boiler, as originally provided is ample, what further contribution can the syphon make toward eliminating the avoidable weight in the design of this locomotive? The original weight of this locomotive was 269,000 lb. If 2250 boiler horsepower is sufficient and the application of syphons enables a reduction in the locomotive weight to 113 pounds per boiler horsepower, then it should be possible to build a syphon-equipped locomotive of this capacity weighing approximately 254,000 lb., which is 15,000 lb. less, or to increase the cylinder horsepower of locomotive No. 50,000 from 2427 to approximately 2600 horsepower by means of thermic syphons, without increasing the weight per axle.

Let us see what thermic syphons have actually contributed toward the elimination of avoidable waste in recent locomotive construction. For an example, consider the mountain-type locomotives, which were constructed by the American Locomotive Company for the Chicago, Rock Island and Pacific R.R. last year. These locomotives are equipped with three syphons each and have a calculated boiler horsepower of 2855. Without syphons, the capacity of these boilers would be reduced to 2550 horsepower. As these locomotives weigh 369,000 lb., their weight with syphons is 129.3 lb. per boiler horsepower, while without syphons, their weight would be increased to 144.7 pounds per boiler horsepower. The cylinder horsepower of these locomotives is 2824 and to have provided a 100 per cent boiler without syphons, on the basis of 144.7 pounds per boiler horsepower, would have increased the total weight of the locomotive to approximately 379,000 lb.

From a statement issued by the Interstate Commerce Commission, it is noted that the cost of these locomotives was approximately $70,000, or 19 cents per pound. At this rate it is apparent that if the railroad had purchased a Mountain-type locomotive having a 100 per cent boiler without syphons, the first cost due to increased weight would not only have been $1900 greater, but the design of this locomotive would have included 10,000 lb. of added weight. Clearly, this is an instance of a most practical avoidance

of waste in locomotive design. Moreover, it is obvious that by employing syphons in these locomotives, the railroads also eliminated avoidable waste in "first cost," besides securing a more efficient locomotive that will continue to eliminate waste of fuel and upkeep every day it is operated.

From the foregoing, it will be concluded that locomotives built or operated without syphons represent avoidable wastes from the standpoints of both fuel economy and unnecessary weight.

An examination of the actual facts is convincing on the point that syphon maintenance is a negligible factor; and that the syphon affords a potential safeguard against disastrous boiler explosions.

JOHN E. MUHLFELD. As the author considers this subject from the constructive standpoint of what constitutes good designing of new locomotives, with particular reference to brick arches, superheaters, feedwater heaters, boiler design and proportions, grate area, mechanical stoking, smokebox draft appliances, cylinder and driving-wheel proportion, static and dynamic weights and forces, and trailing wheel boosters, the writer will mention only the principal factors that have been omitted and which will accomplish much in eliminating otherwise avoidable waste. These are increased boiler pressure and superheat, compounding, reduction in factor of adhesion and tender boosters.

Locomotive boiler working pressures now range from 175 to 205 lb., with a few exceptions of from 210 to 250 lb., in combination with an average of from 150 to 200 deg. fahr. superheat, which in some cases will run as high as 250 deg. fahr. In new designs of locomotive boilers and superheaters the use of 350 lb. of working steam pressure in combination with 300 deg. fahr. superheat, which is now being adopted for stationary boiler service, as compared with 200 lb. steam pressure superheated 300 deg. fahr., will produce for the addition of only 18 B.t.u., or 1.3 per cent of the total heat in the steam and a rise of only 48.5 deg. fahr., or 7.05 per cent in the temperature, *an increase of* 150 *lb. or* 75 *per cent in steam pressure.*

During the period 1890 to 1900 many compound locomotives were put into use in all parts of the United States, these being principally of the two- and four-cylinder types. Unfortunately these double-expansion-cylinder locomotives were built prior to the inauguration of high boiler pressures and superheated steam which resulted in their early conversion to single-expansion-cylinder types due to machinery failures, contributed to largely by cylinder con-

densation. There is no doubt as to the boiler and fuel economy which obtained from the use of these compounds nor of their greater starting power, and in combination with superheated steam and higher boiler pressures, cross-compounding today offers one of the greatest possibilities for improved steam locomotive performance and economy.

With thousands of freight locomotives in successful operation with factors of adhesion, up to a speed of 8 miles per hour, as low as from 3.75 to 4.0 there is opportunity to eliminate the hauling of much useless and expensive dead weight that can and should be made productive. During the past few years many existing locomotives designed with factors of adhesion of from 4.1 to 4.5 have been reduced to from 3.9 to 4.0 by merely increasing the boiler pressures 5 to 20 lb., with resulting greatly increased hauling capacity and economy.

Modern steam locomotives have been much criticized for dead weight haulage in fuel and water supply and tender, which frequently ranges from 75 to 100 tons, or the equivalent to one or two loaded cars. While this dead load is not of such great concern at speed, it becomes somewhat of a burden at starting, at which time, and up to speeds of 15 miles per hour, it can be made useful by a suitable duplex system of engine and tender-wheel propelling mechanism. In this way, and by elimination of many useless leading and trailing engine wheels, the productive power of new, as also of many existing, locomotives can be greatly increased and the bridge and track wheel loads substantially reduced per unit of total engine and tender hauling power.

There are today, on the railroads in the United States, many relatively modern locomotives that are "whiteleaded" or "laid up serviceable" due to lack of rail transportation business. For this reason, and the fact that the purchase of present types of new locomotives offers an insufficient increase in economy to justify the investment, few locomotives are being built. However, with the development of a freight locomotive with high-boiler-pressure, high-superheat, double-expansion, duplex engine and tender power and low factor of adhesion, which, as compared with the present-day freight locomotive of moderate boiler pressure, low-superheat, single-expansion, and dead-load-tender, would have a relatively higher starting and hauling capacity per unit of weight, and could produce by that means and a 35 per cent fuel saving per unit of work performed, a very substantial reduction in "the cost per 1000 gross ton-miles moved," a demand for new to replace the then obsolete

power would be automatically created by the more favarable operating expense ratio to be obtained.

FRANK H. C. COPPUS. The part of the paper under the heading "Fuel Economy" greatly interests the writer, who is not a railroad engineer but who believes that fuel economy in locomotives has more to do with fuel economy than with locomotives, and who has devoted the last fourteen years almost exclusively to the design and application of combustion apparatus.

The report by the United States Geological Survey on "A Superpower System for the Region Between Boston and Washington" has been already so unfavorably criticized that it would seem unnecessary to make further comment. The steam locomotive can be developed to such a high degree of efficiency, that its drawbar horsepower would be so cheap and its capacity increased to such an extent that the electrification of railroads for the sake of economy would be out of the question, at least on a large scale, for some time to come.

The logical order of locomotive development as far as combustion is concerned, should be the following:

1 Mechanical induced draft in the front end
2 Condensing the exhaust steam and carrying condensate to tender
3 Pumping the hot water from the tender through a waste-steam and waste-gas heater into the boiler
4 Under-grate forced draft in ashpan.

Mechanical induced draft in the front end would eliminate the exhaust nozzle and reduce the back pressure to a maximum of 4 lb. Exhaustive tests made on the Santa Fé Railroad showed that an average *added* indicated horsepower of 24 per cent could be obtained by reducing back pressures to 4 lb. This means, among other things, a saving in fuel of 20 per cent.

The condensate to tender the writer estimates would be from 50 to 75 per cent, depending upon the season of the year.

It needs no explanation that the hot water from the tender pumped through a waste-steam heater would enter the waste-gas heater at 220 deg. fahr. This temperature can easily be raised to 300 deg. in the waste-gas heater. Raising the temperature from an average of 60 deg. the year around as it now is to 300 deg. is an undisputed saving of 20 per cent.

Under-grate forced draft in the ashpan in conjunction with induced draft in the front end would create a balanced draft condition which would largely eliminate the loss of unburned coal in sparks and cinders through the stack — estimated to be between 5 and 20 per cent — would stop the inrush of cold air every time the fire door is opened, enable the burning of cheaper grades of coal and, giving unlimited control, automatic in action, and absolute flexibility, would increase the furnace efficiency. All of which will easily result in an average saving of 20 per cent.

From these savings should be deducted the steam used for the operation of the pump and blowers which will reduce them to 18, 18 and 19 per cent, or a net overall saving in fuel of a little over 45 per cent.

It is difficult even to estimate the indirect savings which would result from drafting locomotives mechanically. The increase in the power of compound engines would be enormous. The writer has figures to show that the *added* indicated horsepower to the low-pressure cylinder would be 53 per cent. What will be the saving due to the elimination of double-heading? Then the reclaiming of the water; what effect is this going to have upon the life of the boiler, the time necessary for cleaning, the increased efficiency due to the decrease in scale forming, the time and coal saved by not having to stop to take in water, especially in freight service, etc., and the saving in boiler repairs on account of the elimination of the inrush of cold air into furnace and around water legs.

The increased earning power of locomotives will be enormous. The writer has figures to show that the reduction in back pressure of 8 lb. in a certain locomotive would give an additional revenue for that locomotive of $31,500 per year on an opposing grade of $1\frac{1}{2}$ per cent and $186,060 per year on a level track. These figures are based on the 1912 average rate per ton-mile.

As the railroads use a large percentage of all the coal mined, thousands of cars would be available for other traffic.

Taking it altogether the writer looks forward with confidence to the reduction of the operating expense of the locomotive to an equivalent of 50 per cent of the present coal consumption, as a starter. As a saving of 1 per cent of the fuel used by the locomotives in the United States would mean a saving of $7,500,000 per year, a saving of 50 per cent would give us a figure pleasant to look upon.

The suggested improvements can be attached to all locomotives now in use and at a cost that will pay for themselves inside of a year.

THE AUTHOR wrote that in his paper he had necessarily confined his remarks to indicating proportions which represented recognized good practice in locomotive design and to citing devices which had been tried out and proven in service. The discussion by Messrs. Sperry, Coppus and Muhlfeld, and to some extent, the discussion by Mr. Nicholson, dealt with locomotive improvements in process of development. Mr. Kiesel's remarks concerning the necessary flexibility of empirical formulae met with the author's concurrence. The constant used by the American Locomotive Company was adjusted to 1000 feet piston speed, and was intended to indicate the maximum mean effective pressure obtainable with a conventional valve and valve gear at that speed.

In regard to the coal and water rate comparison, the results quoted for engine No. 50,000 were the average of a number of tests in road service, while the Pennsylvania figures represented the best single test on their testing plant, and in a laboratory test of this kind conditions could be much more readily controlled and more favorable figures secured.

The superiority shown by the Pennsylvania Class K4S Pacific-type locomotive might, however, be realized to a lesser degree in actual road service.

No. 1817

AVOIDABLE WASTE IN CAR OPERATION — THE CONTAINER CAR

By Walter C. Sanders, New York, N. Y.
Associate-Member of the Society

This paper describes the "container" cars which have recently been placed in service on the New York Central Lines.

The advantages claimed for the container car are that less-than-carload lots of freight, mail and express may be shipped with a material saving of time to the shipper and the railroad company, in that the container car may be rapidly unloaded and reloaded. The car will also permit the railroad to secure the maximum mileage out of rolling stock.

Valuable commodities of all kinds can be transported from consignor to consignee inviolate from damage by fire, weather, breakage or theft, also eliminating to a great extent rehandling, trucking and checking.

THE container car was an outcome of the railroad congestion during the war and was first put into operation last year by Mr. A. H. Smith, president of the New York Central Lines, to reduce the transportation losses due to congestion which tied up industry. This congestion was caused by the railroads being unable to get rolling stock near the platforms or places where cars could be unloaded and reloaded and put back into circulation. In some cases manufacturing plants used box cars as a storage warehouse, gladly paying the demurrage charged by the railroads. It is hoped that mail, express and freight robberies, breakage, checking and rehandling, delays to shippers, and many other railroad evils may also be materially reduced by the container-car system.

2 Loss of and damage to freight has grown in recent years into one of the heaviest leaks in the transportation industry and strenuous campaigns which included maintenance of extensive police and supervisory forces, together with educational campaigns among shippers and railroad employees to secure stronger packing,

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careful handling and suppression of theft, have failed to stop this economic waste.

3 The proportions of this transportation problem may be judged from the fact that in the year 1914 American railroads paid out $33,000,000 in claims for loss of and damage to freight, and for the year 1920 this mounted to a total of $125,000,000, the incidental injury to business affected being considerably greater. Under the ordinary system of handling less-than-carload lots or shipments the goods are checked and handled item by item from shipper to truck or dray, from truck to depot platform or warehouse, and from the platform to the car. They are subject to handling and checking at each stage of the journey, and when finally they reach their destination this handling and checking is all done over again. It is therefore necessary to maintain armies



FIG. 1 PASSENGER-EQUIPMENT TYPE OF CONTAINER CAR WITH CONTAINERS IN PLACE

The steel sides of the car prevent the doors of the containers from being opened.

of employees to act as freight handlers, clerks, checkers and station overseers, as well as switchmen to shunt cars to fixed locations where loading and unloading are possible.

4 The container system provides that the portable container shall be loaded and locked at the shipper's own store door, conveyed by motor truck to the railroad yard, and lifted by crane aboard the container car, where steel bulkheads and sides form absolute protection against opening the container in transit. At the destination the locked container is unloaded by a crane and carried by motor truck directly to the warehouse or consignee's door, to be unloaded at his convenience. This simple system of handling goods will make it possible greatly to reduce the force of employees now necessary.

5 Another advantage of the container-car system expected to prove most valuable is the greatly increased use of container rolling stock in moving service, which is particularly important when

traffic expands to its "peak" and the prime need is to shorten layovers of cars in yards and stations for loading and unloading, and to limit their idleness and obstruction through misuse for storage purposes. In busy times the need is to keep every wheel turning as continuously as possible to secure maximum transportation. With ample supplies of the removable containers, which in their several classes are of uniform size and interchangeable,

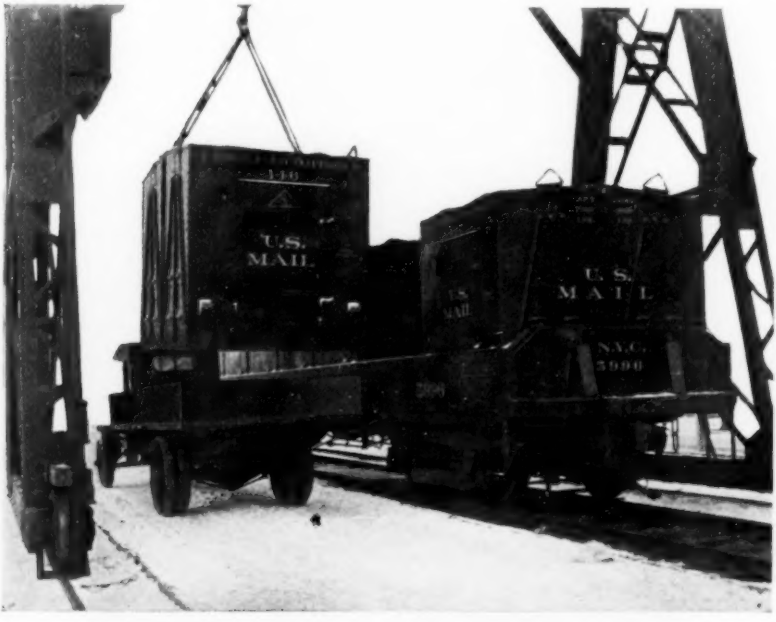


FIG. 2 UNLOADING CONTAINER OF U.S. MAIL TO MOTOR TRUCK FROM PASSENGER-EQUIPMENT TYPE CONTAINER CAR

It will be noted that the sides of motor truck form the same protection against opening of container door as do sides of car.

one carload of containers may be removed and sent with their loads to consignees, and another set immediately hoisted into place and the car be ready to proceed within a matter of minutes in most instances. The locked containers may remain on station platforms or at the stores of shippers for loading or unloading at convenience without tying up costly rolling stock at points where track capacity is limited and congestion quickly obstructs the flow of traffic unless the cars are kept moving. With this rapid handling of the con-

tainers on and off the car the mileage per year made by the ordinary piece of rolling stock may be doubled, and it is predicted that the tremendous expense of maintaining box cars and other rolling stock equal to all emergencies will be materially cut down.



FIG. 3 PASSENGER-EQUIPMENT SERVICE TYPE OF CONTAINER CAR WITH TWO CONTAINERS REMOVED

The guides which hold the containers firmly in place are visible at the sides.

6 The container car may make costly packing and crating unnecessary because goods packed in flimsy pasteboard boxes or even bound with heavy paper are protected against breakage, theft, and water or weather damage.

7 There are at present in service on the New York Central Lines, three container cars one of the mail or express type, and two that are being used for valuable freight, such as silks and woolens. Three new mail cars of improved design which are now being constructed will be equipped with an improved type of all-steel container. A new freight-type container car is being designed, and refrigerator and tank container cars are contemplated.



FIG. 4 HOISTING CONTAINER OF U.S. MAIL FROM PASSENGER-TYPE CONTAINER CAR BY CRANE

The shoes at each corner of container fit between the guides seen in Fig. 3 holding the containers firmly. The operation of moving container from truck to car takes but a minute.

GENERAL DESCRIPTION OF THE CONTAINER CAR

8 The container car is nothing more than a long car with a steel side or fence, similar to a low-side gondola, loaded with large steel safes or containers, made as light as possible, in which commodities of all kinds travel from consignor to consignee, inviolate against thieves, fire, weather and breakage. The safes or containers are lifted on and off the car by cranes or hoisting devices, permitting

the "parent" rolling stock to continue in immediate transportation circulation. In appearance the loaded car seems to be a solid load of safes or steel vaults set aboard a steel underframe with a low protecting steel fence.

EXPRESS CONTAINER CARS

9 The present express car now in use is 61 ft. 3 in. in length, with trucks and fittings that make it interchangeable with standard passenger equipment for use in any passenger train. All modern safety appliances for a special type of car are installed, such as air brakes, hand-hold irons, sill steps, etc. This car carries nine containers, each 9 ft. wide by 6 ft. long, with an inside clear height of 7 ft. 4 in. and a door 3 ft. by 6 ft. The containers are as nearly burglar-, fire-, and weather-proof as it is possible to make them, with a carrying capacity up to 7000 lb. They have wood floors and are arranged with four eyelets or hooks, one at each top outside corner, for convenient lifting and handling. The doors are of standard refrigerator design and are arranged with a hasp and staple for the shipper's padlock, also with pin and slot for standard freight-car seal.

10 The steel sides of the car are spaced to suit the width of the containers, with a clearance of $\frac{1}{2}$ in. on each side; the containers cannot be made as wide as the car itself because of the various state highway laws which regulate the width of motor trucks. The New York State law, for instance, limits the width of an auto truck, including body, to 8 ft., the height from road to top of truck or lading to 12 ft. 6 in., and the truck and lading must not have a combined weight of over 25,000 lb. All container cars are fitted with bulkheads and steel sides 2 ft. high into which the containers fit.

11 In the interior of the body of the gondola part of the car are sectional guides, made of $\frac{1}{4}$-in. steel, resembling channel steel, with the laps projecting out from the open ends to enable them to be riveted to the sides of the car. There are guides at each place on the side of the car where the corner of a container will lie.

12 At each corner on the length side of each container there is a shoe, also of pressed steel, which will fit into the socket formed by the two guides. This arrangement guides the containers to the floor of the car and holds them in place while the car is in transit, without additional blocking or fastening. If it is found necessary to move the car over the road after one or more containers have been

removed, it is unnecessary to put in any kind of blocking, because these guides and shoes are arranged to hold the containers and prevent them from shifting or falling over due to the oscillation of the car.

FREIGHT CONTAINER CARS

13 The two freight-type container cars are 50 ft. long. This is exceptionally long for a car used in freight service, being 9 ft. longer than the longest of New York Central box cars. The general construction is similar to that of the express-type container car.

14 The freight type carries 6 containers, length 6 ft. 7 in., width 7 ft. $1\frac{1}{2}$ in. and clear height 7 ft. $7\frac{3}{8}$ in., all inside measurements.

15 Each container has a steel frame, base, and roof of open-hearth steel, sides or walls of fir wood, and a radial roof so that water will shed off. All joints are double-riveted. The door in the freight type of container is located in the width side and is of standard refrigerator design. As the door is in the end or width side of the container, five 12-in. by $\frac{1}{4}$-in. steel plates are riveted to the sides of the car, extending crosswise the inside of the car, through the open clearance spaces between containers, so in case one or more containers are removed from the car, the doors of the remaining containers cannot be opened.

CONTAINERS AND CONTAINER CARS NOW UNDER CONSTRUCTION

16 The three new mail-type container cars now being built will carry 8 containers of a new, improved design, the outside measurements being, length 7 ft. $2\frac{1}{2}$ in., width 9 ft. $3\frac{1}{2}$ in., clear height 8 ft. 2 in., with a 5 ft. 9-in. by 3 ft. 6-in. door on the length side of the container. The cubic capacity will be 438 ft., light weight 3000 lb. and capacity 7000 lb.

17 Tests conducted in the last few months on the New York Central Lines have demonstrated that the express type of container car can be emptied of the nine containers by an ordinary crane in 21 min. and reloaded with other containers and the car put back in circulation in about the same time. This test was made with an ordinary moving track crane, since no special cranes or handling devices have as yet been constructed for use in handling the containers. With the special handling devices contemplated it will be possible to unload the containers with greater speed directly to waiting motor trucks, platforms, or on the ground.

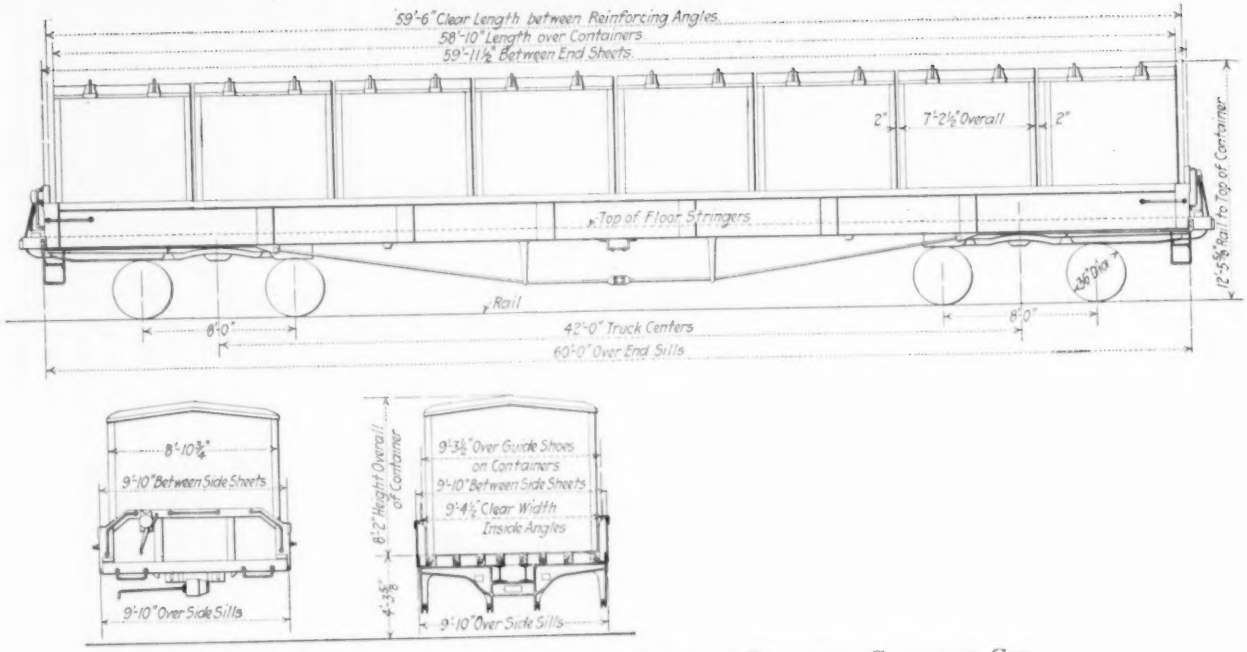


FIG. 5 GENERAL ARRANGEMENT OF PASSENGER-EQUIPMENT CONTAINER CAR

This car carries 8 containers.

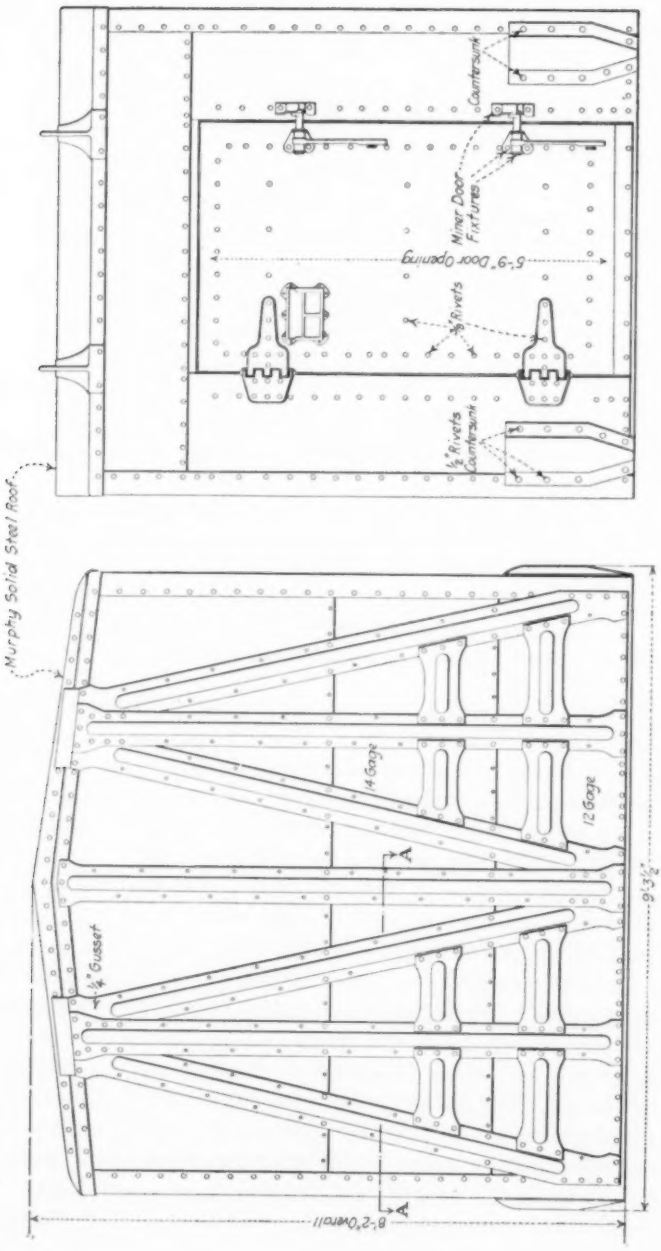


FIG. 6 GENERAL ARRANGEMENT OF THE NEW IMPROVED TYPE OF MAIL CONTAINER

Note the eyelets for lifting and handling at the top.

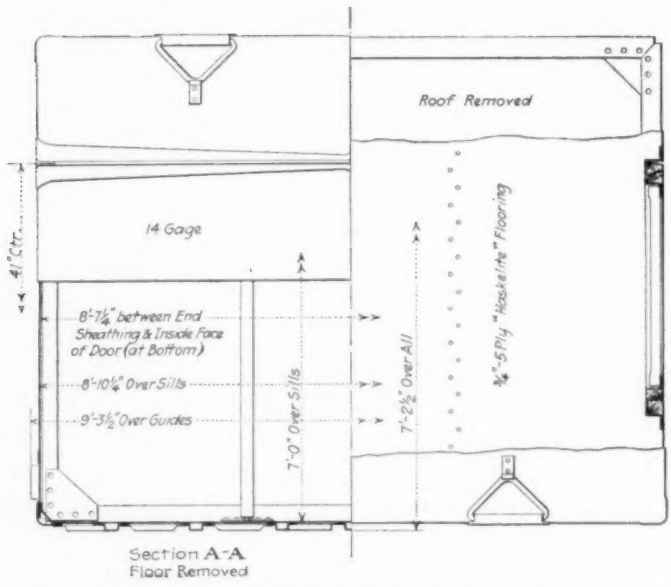


FIG. 6 GENERAL ARRANGEMENT OF THE NEW IMPROVED TYPE OF MAIL CONTAINER

Note the eyelets for lifting and handling at the top.

18 During May 1921, at the request of the Postmaster General, a mail test was run from New York City to Chicago, Ill., and return with the express type of container car. Upon arrival at Chicago the nine containers, containing 37,000 lb. of mail, were unloaded on to waiting Post Office motor trucks in 21 min., which is one-fifth of the time used in unloading an ordinary mail storage car. At Chicago connections were made with western mail trains that have never been made before. Upon the arrival of the car at New York on the return trip the containers were removed from the car in 18 min.

19 It is believed that the use of containers in mail service (1) will prevent the loss in transit of valuable registered mail, parcels post and other mail; (2) will mean a saving to the Government in handling mail, both in trucking and checking, as well as a material saving to the railroads in the use of equipment; (3) will make possible a quick transfer at important gateway points and the maintaining of close railway connections otherwise impossible; and (4) will afford an increased weight and capacity as compared with the average load now handled in mail storage or baggage cars, the aver-

age weight of mail now carried in mail storage cars being approximately 30,000 lb.; 37,000 lb. of mail were carried on the run to Chicago.

SUMMARY

20 A summary of the prime advantages of the container-car system is as follows:

(*a*) It will furnish a means of expediting delivery of less-than-carload lots of commodities by eliminating the time and expense of rehandling, checking and trucking

(*b*) It will eliminate costly crating and packing

(*c*) The immediate unloading and loading of containers at terminal points eliminates the item of demurrage, at the same time promptly releasing rolling stock, clearing the yards of cars and reducing congestion

(*d*) It will eliminate the piecemeal loading of cars at railway sidings in exposure to all kinds of weather

(*e*) It will tend to keep the car moving at all times, making possible double the mileage as made now by an ordinary piece of rolling stock

(*f*) Containers are fire- and weather-proof, and also burglar-proof in that they cannot be opened while on the car or while being transferred by handling device to and from the car.

21 The increased service capacity of each unit by the development of the container-car system is thought to hold far-reaching economic possibilities in railroad operation of the future, as well as in the coördinated use of the motor truck and the electric railway.

DISCUSSION

J. N. Crocker.[1] In Par. 18 it is stated that the load carried was 37,000 lb. It should be understood that this was not the indicated weight-carrying capacity of the car for that kind of traffic, for the reason that the mail was loaded into the car as it came from the post office by truck and included parcel post, heavy mail and periodicals. Mail in one container weighed 5280 lb. while mail in others weighed approximately 5000 lb., indicating weight-carrying capacity of approximately 48,000 lb. The Post Office Department

[1] Supervisor of Mail Traffic, New York Central Railroad, Grand Central Terminal, New York.

has authorized the regular use of a line of cars between New York and Chicago commencing January 15, which will handle all kinds of mail. The new cars will have a weight-carrying capacity of 56,000 lb.

This type of car should be especially attractive to the Post Office Department as it makes possible the movement of valuable parcels post, registered mail, money, silk, etc., from the door of one post office to the door of another in what is substantially a burglar-proof vault. In addition it is possible to maintain separations en route, providing quicker transfer at the terminal and at the same time not reduce the weight or cubic foot capacity of the car.

C. H. Otis.[1] The Post Office Department feels that the use of the container car will reduce the opportunity, and when used, will be the means of preventing robbery. This type of car will expedite the dispatch of bulk mail. During the recent test, one container with mail all for the Santa Fé line, was unloaded within three minutes after arrival from New York, container moved over to the Santa Fé station and mail loaded into the mail cars, making a connection never made before for bulk mail.

Numerous cars or parts of cars of parcel post mail have been destroyed by fire. If a fire should start in the container car, it would be confined to the container in which the fire originated, thus preventing its spread, and thereby reducing the extent of the damage.

F. S. Gallagher.[2] With reference to the statement by the author of unloading 37,000 lb. of mail in twenty-one and one-half minutes it should be remembered that 50,000 lb. could have been unloaded as quickly. The container put in service on the New York Central lines is so arranged that the door cannot be opened until the container is lifted off the car or truck. The idea is to make it impossible to open the doors while the container is on its way between the shipper and the consignee. In reference to tank containers mentioned by the author this matter is now pending with several milk companies with the idea of utilizing this container system for the transportation of milk.

In reference to a question by J. C. McBride regarding difficulties with ice Mr. Gallagher said that the cars had been operated from January of last year without trouble and no trouble was ex-

[1] Assistant Superintendent Railway Mail Service, in charge of postal car construction.

[2] Engr. of Rolling Stock, N. Y. C. R. R. Co.

pected because the floors of the new cars would be covered with a wire mesh so that any water would run through the bottom of the car.

A. E. OSTRANDER. Special containers for use in the transportation of high-grade freight are not new; they have been in successful operation for years. One large company in New York has purchased several lots of containers called "steel lift vans," and their method of operation is to supply one of the sevans to a shipper, furnishing him with keys to special locks with which the containers are fitted. The shipper then loads his freight into the container, the lading in this case being high-grade furniture, and locks it up; it is then delivered to the railroad for shipment to any part of the world.

The fact that they have continued in business, and, as indicated above, have purchased several orders of the vans, shows that it must have been profitable business for them; therefore it would certainly seem to be more profitable to a railroad to handle strictly high-grade freight in this manner.

The container car is simply a specially designed railway vehicle adapted to the economical transportation of such a container, and the arguments for or against the container car are really arguments for or against the container method of shipping freight.

In addition to the advantages to the railroad, which the author has so ably brought out in his paper, there is one class of freight which would be highly remunerative if handled by the container method, namely, raw silk and finished silk products. This class of freight at the present time is being handled almost entirely by motor-truck companies owing to the danger of loss and theft, and consequently carries high insurance rates.

Containers might be used profitably by the railroads in the handling of way-freight; of course it would be necessary to have loading platforms at each station, equipped with small cranes to handle the containers on and off the cars; this would involve train initial expense, but considering the time it takes a way-freight train to go over a division as compared with the running time of a through-freight, it is very easy to see the amount of time wasted, or rather, that could be saved if the switching incident to setting out and picking up cars at each station could be eliminated. Furthermore, as the way-freight shipments are usually small, a freight container car of the type described carrying these containers would probably serve for six different stations, whereas under the present method

it is doubtful if the same amount of freight would load into any one box car.

An advantage of the container from the viewpoint of construction lies in the fact that they are separate units, each with small, separate roofs, therefore one of the most serious causes of damage to freight in transit, i.e., leaky roofs, has been reduced to a minimum by the fact that each container has a separate and independent roof, making, in the full car equipment, a series of six small roofs to a car, instead of one. This tendency to roof damage is further reduced by the fact that the container is independent of the car structure and any buckling of the underframe does not affect the roof structure of the container.

Under the description of freight container car, the statement is made that they are equipped with steel plates as an integral part of the car, extending crosswise inside of the car so as to prevent opening of the container doors in case the car is not fully loaded. To the writer this would not appear desirable unless it was determined that the container car was never to be used for any other service. He believes it is very possible, and very desirable, to have such locking arrangements made a part of the container, which could easily be arranged so that, when the container is set in place, the doors could not be opened until the container had been raised a certain distance, say a sufficient height to clear the sides of the car, and with such an arrangement there is nothing to prevent making the container car adaptable to any class of material commonly loaded in a low-side gondola. The reason for this is obvious: there are certain classes of freight which the most optimistic supporter of the container idea will admit cannot be handled in a container, and if the container is to come into very general use, then any gondola, or possibly even any flat car, could be so arranged that containers could be loaded thereon.

In regard to the statement made that the use of the container car saves demurrage on freight cars, this seems unfair to the railroad if no charge is made for demurrage on the containers themselves. In other words, the shipper or the consignee has no more right to use a container as a storage warehouse than he has to hold up a car for the same purpose when he is not in a position to unload his goods.

In conclusion, the writer believes that the container car idea has a bright future and that a solution of the problems, which of course, as with any other device of merit, will take some time to

solve, can be accomplished by close coöperation between the shipper, the transportation department and the mechanical department.

THE AUTHOR. Experimental use of container type of car in handling U. S. mail since January 20, 1922, has confirmed claims originally made for this type of car as to advantages to the Post Office Department, in connection with the transportation of mail as follows:

a There is increased carrying capacity, both in weight and volume, as compared with ordinary baggage or mail storage car.

b Possibility of damage or loss of mails by fire is virtually eliminated for the reason that containers are practically airtight, necessarily confining loss to one container, or one-eighth of the car.

c With this type of car it has proved feasible to handle all classes of parcel post mails outside of sacks, which reduces handling to the minimum, and consequently possibility of damage is materially reduced.

d This type of car eliminates possibility of depredation and loss of mail while in the custody of other than Government employees, for the reason that it is at all times enclosed in steel containers, under Government lock, while being handled and transported between post office at point of origin and destination. This additional protection is especially desirable in the handling of registered, insured and other valuable mails.

e The possibility of prompt handling and transfer of mails at important gateway points is an especially attractive feature. For example, during recent handling at Chicago all eight containers were unloaded from a car, placed on trucks and the trucks moved from the unloading point within eight minutes after car was placed, as compared with 1 to 2 hours allowed for handling mails from a fully loaded mail-storage car. At the same time the container type of car automatically maintains, if necessary, eight separations of the mails without in any way decreasing the weight or cubic capacity of load, thus eliminating rehandling now necessary when mails are carried in an ordinary car.

f The container system of handling mails materially reduces the man-power necessary for both the Government and the railroad to provide in connection with the transportation of the mails.

No. 1818

PROCESS CHARTS

FIRST STEPS IN FINDING THE ONE BEST WAY TO DO WORK

By Frank B. Gilbreth, Montclair, N. J.
Member of the Society

and

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Non-Member

The process chart is a device for visualizing a process as a means of improving it. Every detail of a process is more or less affected by every other detail; therefore the entire process must be presented in such form that it can be visualized all at once before any changes are made in any of its subdivisions. In any subdivision of the process under examination, any changes made without due consideration of all the decisions and all the motions that precede and follow that subdivision will often be found unsuited to the ultimate plan of operation.

In the present paper the authors point out the place of the process chart in management and present established working data used successfully in numerous working installations for many years. They also point out its simplicity, field of application, its relation to standardization, etc., etc.

While the process-chart methods will be helpful in any kind of work and under all forms of management, the best results can come, the authors state, only where there is a mechanism of management that will enforce and make repetitive the conditions of the standards.

THE Process Chart is a device for visualizing a process as a means of improving it. Every detail of a process is more or less affected by every other detail; therefore the entire process must be presented in such form that it can be visualized all at once before any changes are made in any of its subdivisions. In any subdivision of the process under examination, any changes made without due consideration of *all* the decisions and all the motions that precede and follow that subdivision will often be found unsuited to the ultimate plan of operation.

Presented at the Annual Meeting, December 1921, of The American Society of Mechanical Engineers.

PLACE OF PROCESS CHART IN MANAGEMENT

2 The process chart is a record of present conditions. It presents, in simple, easily understood, compact form, data which must be collected and examined before any improvement in existing conditions and methods is undertaken. Even if existing conditions are apparently satisfactory, the chart is useful as presenting much information in condensed form.

3 The process chart serves as an indicator of profitable changes. It assists in preventing "inventing downward," and stimulates invention that is cumulative and of permanent value. It is not only the first step in visualizing the *one best way to do work*, but is useful in every stage of deriving it.

4 This paper presents established working data used successfully in numerous installations for many years.

FIELD OF APPLICATION

5 The process chart lends itself equally well to the routine of production, selling, accounting and finance. It presents both simple and complicated problems easily and successfully; it provides records that are comparable; it assists in solving problems of notification and interdepartmental discrepancies, and it makes possible the more efficient utilization of similarities in different kinds of work and in the transfer of skill.

6 During the stress of unexpected rush in production, it is often considered advisable to continue existing practice in present processes, even though inefficient. On the other hand, when production is normal or slow, it is more generally conceded that processes can profitably be bettered.

7 The use of this process-chart procedure permits recording the existing and proposed methods and changes without the slightest fear of disturbing or disrupting the actual work itself, and also regardless of whether business conditions are usual or unusual.

8 Those who are interested in improving their processes of production should utilize times of industrial depression for that purpose. Many concerns are now taking such action; many more could undoubtedly enter upon such procedure of scrutinizing all their processes with the idea of putting them in the best possible condition, if they knew a simple procedure of such analysis.

SIMPLICITY OF THE PROCESS CHART

9 The aim of the process chart is to present information regarding existing and proposed processes in such simple form that such information can become available to and usable by the greatest possible number of people in an organization before any changes whatever are actually made, so that the special knowledge and suggestions of those in positions of minor importance can be fully utilized.

10 The time has passed — if it ever existed — when the engineer prided himself upon the abstruse material that he studied and

136. All the time of ALL our workmen must be entered on the regular time book sheets. When men are employed on EXTRA WORK see additional directions below.
137. Time book sheets must be mailed to the office the night that the payroll closes. Send at the same time all EXPENSE ACCOUNTS, CASH ACCOUNTS, (made up to date) and PAYMASTERS' REQUISITION.
138. Send original time sheet to the office regardless of its appearance.
139. Keep all time books with the greatest care.
140. DO NOT USE A NOTE BOOK. We find that most mistakes in workmen's time are caused by copying.
141. All remarks about the payroll must be written on the time sheet itself.
142. The full time of every one of our men must appear on the regular TIME BOOK ("TB") sheets. The time of any of workmen which is to be charged to EXTRA WORK must be entered on supplementary sheets under the proper ORDER LETTER number.
143. Keep separate sheet for each ORDER LETTER and fill out carefully the ORD. LET. (NUMBER) AND NATURE OF WORK.

FIG. 1 PORTION OF PAGE OF WRITTEN SYSTEM REPRODUCED DIRECTLY FROM TYPEWRITTEN SHEETS

presented. Today engineering ranks with the other sciences in conveying ideas in a form that is immediately usable. We avoid "translating," interpreting and adapting, thus eliminating waste.

11 The process chart has met the tests of a satisfactory teaching device from the psychological standpoint, as well as of a satisfactory working device from an engineering standpoint. It shows the planned process as well as the present process, and therefore gains the coöperation of those affected. In many instances recording industrial processes in process-chart form has resulted in aston ishing improvements.

COLLECTING THE INFORMATION

12 Process-chart notes and information should be collected and set down in sketch form by a highly intelligent man, preferably with

an engineering training and experience, but who need not necessarily have been previously familiar with the actual details of the processes. In fact, the unbiased eye of an intelligent and experienced process-chart maker usually brings better results than does the study of a less keen man with more special information regarding present practices of the processes. The mere act of investigating sufficiently to make the notes in good enough condition for the draftsman to copy invariably results in many ideas and suggestions for improvement, and all of these suggestions, good and bad, should be retained and filed together with the description of the process chart. These suggestions and proposed improvements must be later explained to others, such as boards of directors, managers and foremen, and for best results also to certain workmen and clerks who have special craft or process knowledge. To overcome the obstacles due to habit, worship of tradition and prejudice, the more intelligence shown by the process-chart recorder, the sooner hearty coöperation of all concerned will be secured. Any one can make this form of process chart with no previous experience in making such charts, but the more experience one has in making them, the more certain standard combinations of operations, inspection and transporting can be transferred bodily to advantage to the charts of proposed processes.

UTILIZING SUGGESTIONS

13 A new viewpoint concerning old conditions invariably comes to those members of the organization who have become so accustomed to the traditional method that they cannot easily visualize a new method without prejudice until they actually see it in a new graphical form. After the rough notes of the process-chart maker have been redrawn and blueprinted, they are later exhibited in the executives' theater.

14 If discussions arise as to the correctness of the presentation of the existing facts, or as to further details of the operations being studied, as shown by the simple symbols of the process chart, the room can be darkened and inexpensive glass diapositives projected on the wall. In addition, those present may be supplied with a special small pocket folding stereoscope for use with the same glass diapositives.

15 As soon as the old or existing process is understood, a process chart of a better sequence and kind of operations which compose it is made. The procedure for this is the same for all cases as far as

they are carried for the time being, but of course those processes which warrant the most study should be carried farthest in the



FIG. 2 STANDARD CHANGE ORDER BLANK

Note that all blank forms should be thus numbered in each blank space to be filled out for describing clearly the One Best Way to Learn Work.

process-chart procedure. The more people who see the process chart and the greater detail into which the regular process charts are divided, the more suggestions for improvement will come in.

RELATION TO STANDARDIZATION

16 There is no process that warrants a process chart that does not warrant a "write-up" or "written system." Fig. 1 shows a portion of a page of the written system of our organization printed in 1899. It is interesting to note that it was reproduced directly from typewritten sheets and without being set up in type, and it is believed to be the first instance on record of a book being printed from zinc etchings made from typewritten manuscript.

17 The more care taken in making the written system, the more will develop the need for and appreciation of the value of clearly defined written standards. The better and the more detail in which the written system is developed, the better and easier will the standards and standing orders be developed.

18 Standards in writing should be made, even if there is not the managerial mechanism necessary to enforce and maintain them. Standards made even with enforcing mechanism absent will hasten the day when the enforcing and maintaining mech-



FIG. 3 STANDARD SYMBOLS FOR PROCESS CHARTS

(Continued on page 9)

anism will be installed and continuously operated. The procedure of making the standards will invariably lead to the simplifying and improving of the various steps as shown on the process chart.

19 If it is desirable to study, improve and still further identify the subject-matter of each part of the process chart, it should be submitted to the regular routine process of standardization. A standard is a matter of degree. In its best form it is identified and defined with all the care and precision of the best practice for making the standing orders. The range, however, is dependent upon the degree of perfection with which provision has been made for enforcing and maintaining standards.

20 While on the subject of range, it is well to call attention to the remarkable attempts of Germany and Holland to provide national standards. These standards already cover a very wide field, from the style of the hand lettering and the rulings to be used on the paper on which the standards themselves are printed, to a metal seat for a harvester, tractor or tank. The range, in fact, already covers a surprisingly wide list of things which have not been properly standardized in America, and is intended eventually to cover everything that is manufactured in quantity, or that will for any other reason reduce costs or improve quality. Although there is much to criticise in these foreign standards, they are highly meritorious, worthy of continuous and careful attention, and a great credit to those who have devised them.



FIG. 3 STANDARD SYMBOLS FOR PROCESS CHARTS (*Continued*)

21 It must be remembered that the kind of standard adopted will affect the process almost invariably. Therefore standardization must be considered if the *one best way to do work* is to be derived.

22 Particular attention should be called to the fact that the creation of national standards of manufacture, even to the smallest components of the arts and trades, means also the stabilization of employment and business in general, because manufacturers without sufficient orders in their regular lines of business to keep going will find it more profitable, in many instances, to manufacture the national standards and thus to turn their stores inventories into money immediately, rather than let their specially trained and skilled men leave them, with all the disadvantages of a high labor turnover. Here is an endless spiral of benefit, for the more chances there are for a manufacturer to dispose of his inventory for cash and keep his organization together a little longer, even in times of general timidity, the more he will dare be a purchaser of raw material, for the process for such emergencies can be standardized and ready. The result is standardization combined with stabilization of employment, a quick capital turnover and a low labor turnover.

23 Many fear standardization of the component elements of a process chart as something from which, once done, it will be difficult to escape. For the purpose of allaying such groundless fears, the standard change order, Fig. 2, has been provided. This, when signed by the authorized party, instantly changes, or for a certain instance, or a certain time, waives the existing standard whether it relates to a thing, a method, a procedure or a process. It will be noted that this change order blank contains provisions for the notification of, and for the acknowledgment of receipt of notification of, all persons who are concerned with, or interested in, the change.

24 Note that in the lower right-hand corners of the various spaces in Fig. 2 there are small consecutive numbers. This is standardized to agree with write-ups and standing orders for using standard blank forms. It not only makes the writing of the standing order more simple, exact and clear, but it also shortens the time of the learning period for using these blank forms. This is a valuable feature at all times, but particularly useful during the transitory period of installing new methods of management.

25 Experience shows that if process charts are made use of, exceedingly few of the existing blank forms survive in their present form. The savings that can be made in any large organization resulting from submitting them to the test of this process will in variably prove it to be a good investment.

26 If all departments of the United States Government would adopt two features, namely:

a Put small numbers in each space to be filled out on all of its blank forms, and

b Make write-ups and standing orders of exactly how each blank form is to be filled out

and would then make a survey and criticism in accordance with the known laws of micromotion study, the resulting savings would be astounding.

27 We believe that, as a result, not one per cent of present

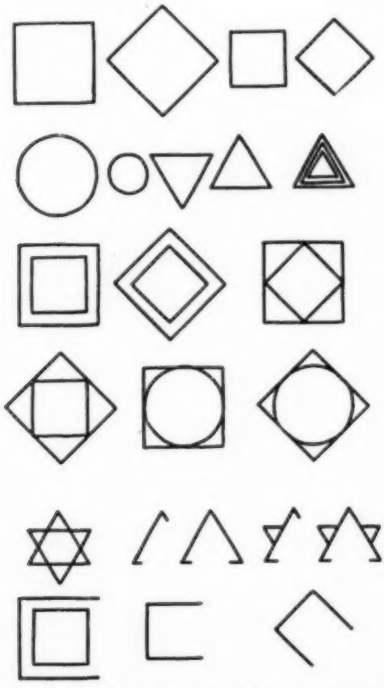
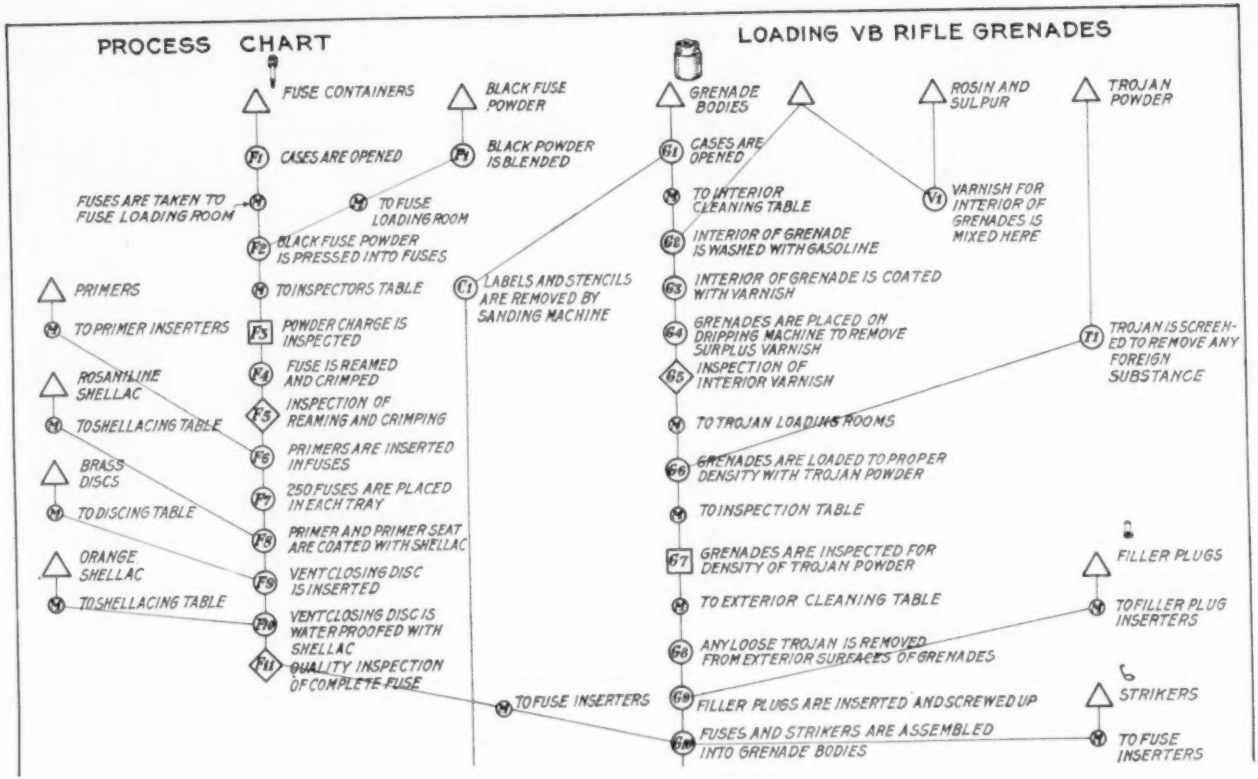


FIG. 4 STANDARD PROCESS-CHART SYMBOLS MADE WITH STANDARD CELLULOID GUIDES AND USING STANDARD PEN NO. 707

(One-fourth actual size)

blank forms would remain unchanged. All Government blank forms that we have seen violate all laws of motion study and learning methods of least waste.

28 The standing order is for enforcing standards and other standing orders. This has already been described in a paper before this Society.


PROCESS CHART
LOADING VB RIFLE GRENADES
FUSE CONTAINERS
BLACK FUSE POWDER
F1 CASES ARE OPENED
P1 BLACK POWDER IS BLENDED
FUSES ARE TAKEN TO FUSE LOADING ROOM
M TO FUSE LOADING ROOM
F2 BLACK FUSE POWDER IS PRESSED INTO FUSES
M TO INSPECTORS TABLE
PRIMERS
M TO PRIMER INSERTERS
F3 POWDER CHARGE IS INSPECTED
F4 FUSE IS REAMED AND CRIMPED
ROSANILINE SHELLAC
M TO SHELLACING TABLE
F5 INSPECTION OF REAMING AND CRIMPING
F6 PRIMERS ARE INSERTED IN FUSES
BRASS DISCS
M TO DISCING TABLE
F7 250 FUSES ARE PLACED IN EACH TRAY
F8 PRIMER AND PRIMER SEAT ARE COATED WITH SHELLAC
ORANGE SHELLAC
M TO SHELLACING TABLE
F9 VENT CLOSING DISC IS INSERTED
F10 VENT CLOSING DISC IS WATER PROOFED WITH SHELLAC
F11 QUALITY INSPECTION OF COMPLETE FUSE
M TO FUSE INSERTERS
C1 LABELS AND STENCILS ARE REMOVED BY SANDING MACHINE
GRENADE BODIES
G1 CASES ARE OPENED
M TO INTERIOR CLEANING TABLE
G2 INTERIOR OF GRENADE IS WASHED WITH GASOLINE
G3 INTERIOR OF GRENADE IS COATED WITH VARNISH
G4 GRENADES ARE PLACED ON DRIPPING MACHINE TO REMOVE SURPLUS VARNISH
G5 INSPECTION OF INTERIOR VARNISH
M TO TROJAN LOADING ROOMS
G6 GRENADES ARE LOADED TO PROPER DENSITY WITH TROJAN POWDER
M TO INSPECTION TABLE
G7 GRENADES ARE INSPECTED FOR DENSITY OF TROJAN POWDER
M TO EXTERIOR CLEANING TABLE
G8 ANY LOOSE TROJAN IS REMOVED FROM EXTERIOR SURFACES OF GRENADES
G9 FILLER PLUGS ARE INSERTED AND SCREWED UP
G10 FUSES AND STRIKERS ARE ASSEMBLED INTO GRENADE BODIES
ROSIN AND SULPUR
V1 VARNISH FOR INTERIOR OF GRENADES IS MIXED HERE
TROJAN POWDER
T1 TROJAN IS SCREENED TO REMOVE ANY FOREIGN SUBSTANCE
FILLER PLUGS
M TO FILLER PLUG INSERTERS
STRIKERS
M TO FUSE INSERTERS

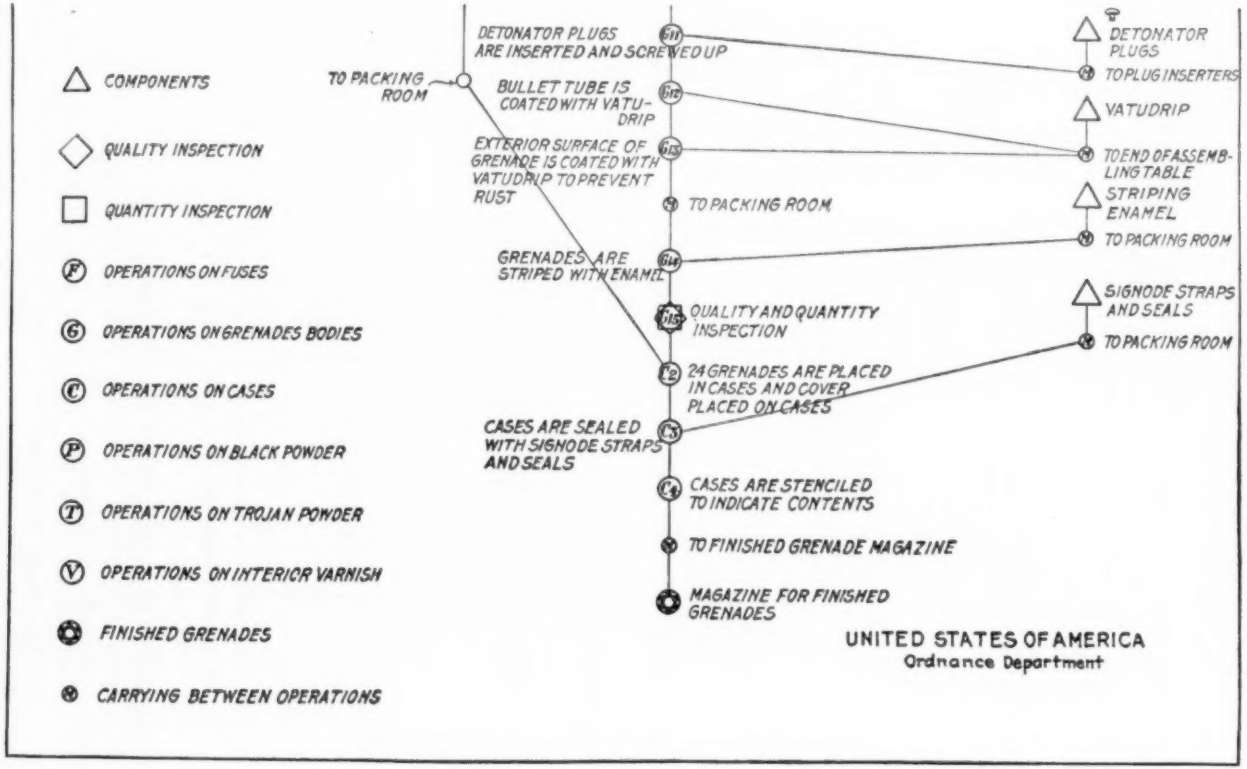


FIG. 5 PROCESS CHART FOR LOADING RIFLE GRENADES

29 The more detail in which the standing order is made, the better. The more the procedure is described by it, the greater will be the improvements and the greater the automaticity resulting.[1]

30 If any operation of the process shown in the process chart is one that will sufficiently affect similar work, then motion study

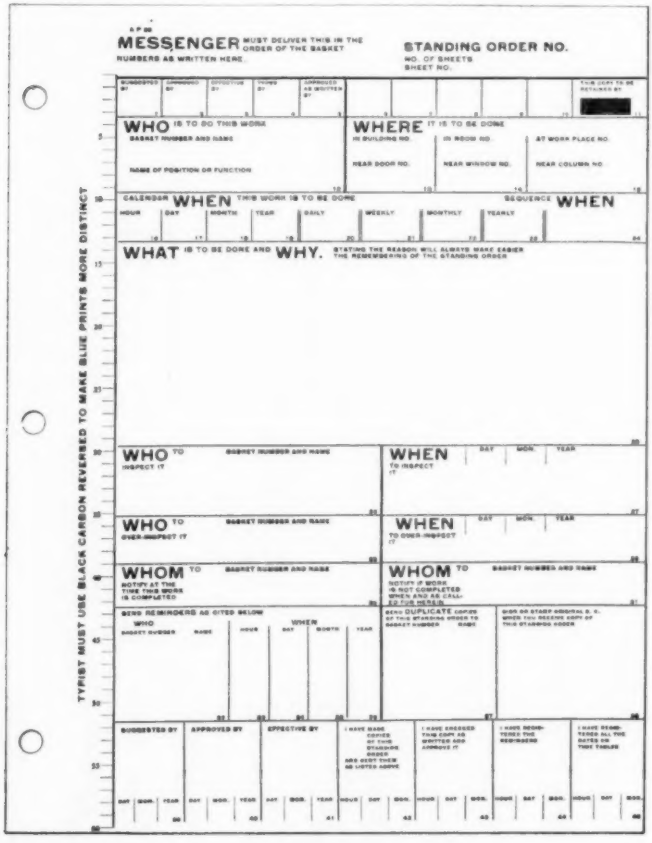


FIG. 6 STANDING ORDER BLANK

should be made of each part of the process, and the degree to which the motion study should be carried depends upon the opportunities existing therein for savings.

[1] See Psychology of Management; Applied Motion Study; and *Bulletin of the Taylor Society* for June, 1921.

31 If the operations are highly repetitive or consist of parts or subdivisions that can be transferred to the study of many other operations, then micromotion studies already made can be referred to; also new and further micromotion studies may be warranted in order that the details of method with the exact times of each of the individual subdivisions of the cycle of motions, or "therbligs," as they are called, that compose the one best way known, may be recorded for constant and cumulative improvement. Such motion study can be best visualized if seen in chart form and similar process charts can be made of any or all of the large or small circles, squares and diamonds shown on the process charts. These subdivided motion charts can be made of each and all of the cycles in any given operation. Much benefit can often be derived, even if such motion charts are made roughly. For best results, and especially when complete records are required, such, for example, as when the process charts are of work that is highly repetitive, micromotion charts can be made which will give the maximum amount of analysis and visualization of component parts of the existing

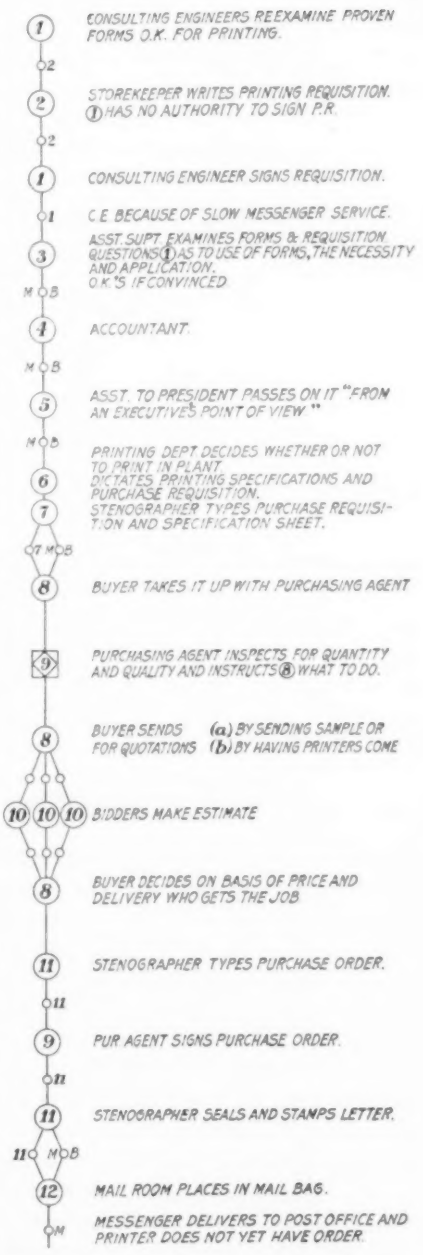


FIG. 7 PROCESS CHART FOR ORDERING BLANK FORMS — PRESENT METHOD

and proposed processes. These can be still further visualized by the chronocyclegraph processes. Both the chronocyclegraph and the micromotion process have been described before the Society and more recent developments in these methods and devices for visualizing existing and proposed processes will be the subject of a later paper.

32 The records of the micromotion study and the chronocyclegraph methods and devices present permanently all the facts in such form that they can be used at any time. These photographic records can be studied as slowly as desired, regardless of how fast the motions of the process were actually made and the marvels of the details of



FIG. 8 PROPOSED PROCESS CHART FOR FIRST ORDERS

superskill, unknown and unrecognized even by those who possess it, can be studied at will, leisurely and intensively, by learners everywhere, far as well as near. If desired, these errorless records may be used only as far as to fill the need of present requirements, or they may be laid away until further needs demand further study, such records being in such perfect detail that they are practically as usable when old as when new. These permanent records of complete sequences of details of complete processes furnish the foundation of the best kind of trade and industrial education, namely, the dissemination of detailed instructions as to the synthesized processes of the best workers obtainable.

33 These synthesized records of details of processes in turn may be further combined and large units of standard practice become available for the synthesis of complete operations in process charts.

34 While the process-chart methods will be helpful in any kind of work and under all forms of management, the best results can come only where there is a mechanism of management that will enforce and make repetitive the conditions of the standards.

MECHANISM OF MAKING PROCESS CHARTS

35 There are shown herewith:

a The symbols used with their meanings (Fig. 3)
b The mechanical devices for making the symbols on the process charts (Fig. 4)
c Completed process chart (Fig. 5)
d Accompanying forms (Fig. 6)
e Illustrations of collecting and using data.

SUMMARY

36 The procedure for making, examining and improving a process is, therefore, preferably as follows:

a Examine process and record with rough notes and stereoscopic diapositives the existing process in detail

b Have draftsman copy rough notes in form for blueprinting, photographic projection and exhibition to executives and others

c Show the diapositives with stereoscope and lantern slides of process charts in executives' theater to executives and workers

d Improve present methods by the use of —

1 Suggestion system
2 Written description of new methods or "write-ups," "manuals," "codes," "written systems," as they are variously called
3 Standards
4 Standing orders
5 Motion study
6 Micromotion studies and chronocyclegraphs for obtaining and recording the One Best Way to do Work.

e Make process chart of the process as finally adopted as a base for still further and cumulative improvement.

37 Note that —

a Visualizing processes does not necessarily mean changing the processes

b Process charts pay.

DISCUSSION

FRED H. COLVIN. The one great trouble with most records is that they are usually so delayed as to become merely post-mortems after the job is finished and are not even always of value for the next job.

The process chart has many advantages, particularly as it can be readily referred to and easily visualized. The desire to secure information "regarding existing and proposed processes, in such a simple form that it can become valuable to and used by the greatest possible number of people in the organization before any changes whatever are actually made," shows that the authors realize one of the greatest obstacles in the way of introducing new methods. It is not easy for either executives or workers to change their methods and habits. And, when the changes are ordered without consultation and must be followed simply because they are autocratically ordered from above, a natural resentment arises which makes any change difficult for all concerned. An understanding of this phase of human nature, which exists regardless of the position held, is the first step toward efficient management.

Utilization of photography in this work has opened up new possibilities not only of studying motion but of presenting the results of this and other studies in such a way as to be readily understood in the various departments of industry. Utilizing this information in the preparation of process charts makes them of special value.

The question of standards and standardization which are referred to in this paper are also well worth serious consideration. For while it is of course easy to carry standardization too far, or more correctly perhaps to standardize the wrong things, there can be no question as to the economic advantage of considerable standardization along many lines. This, as pointed out in the paper, would aid in the stabilization of employment and of business, both of which are highly desirable from every point of view. It is time for engineers to pay more attention to the economic and business side of industry.

ROBERT T. KENT. The process chart is simply an instruction card on a larger scale. It is a question of applying the instruction card not to the individual operation but to a complete process. It is of the utmost value to every manufacturing organization in which methods are to be fixed. Industrial concerns insure

their plants against fire and insure the lives of their executives, but how many concerns insure their business and manufacturing experience; that is, make permanent the knowledge of how work is done in the plant? This can be done by means of instruction cards and process charts. If there is an instruction card for every operation and a process chart for every process, the entire organization can be removed, and a new organization, with this information at hand, will be able to carry on the business.

MORRIS L. COOKE. The paper is interesting as being further evidence that we are just at the beginning of graphics and symbolization in industry. In these management questions we should start with the present state of the art and show the relation between this development and those of the past. The writer would like to know how Mr. Gilbreth has coördinated this type of chart with what has been done before.

WALLACE CLARK. In installing management methods it has been the writer's practice to have the foreman write down the operations through which the material has gone in his department. The mere fact that he writes them down makes him concentrate on them and if these processes are not in accordance with the best knowledge he has he will make recommendations. What he has written down is shown to the other executives and they focus their knowledge on the methods used. There is no better way for a manufacturing plant to make use of the knowledge it already has than through written records of processes such as the author has shown.

WALTER N. POLAKOV. The paper is interesting and stimulating in this way that it presents graphically the basis of planning and analyzing things which can not be seen in any other way, not even with the stop watch, and while these studies are being made, a live interest in the work is shown, especially if the workers' coöperation is invited. The point to guard against is that as a certain method is established, however good it may be it becomes crystallized, deadly, fixed and monotonous routine so that a man falls into a rut and has no chance to get out of it.

A. B. SEGUR. It seems to the writer that the Society owes to Mr. Gilbreth a special note of thanks for bringing to its attention a simplified method of constructing a process chart, or as it is known in some places, "progress chart."

The method of using symbols, each of which denotes some

special element of manufacturing operation, is apparently old to Mr. Gilbreth. The method is rather new to the writer and greatly simplifies the usual chart. The outline chart used as a key to the individual operation chart is very useful in studying manufacturing layouts.

In making studies of the manufacturing operations of various concerns, we have made use of the progress chart for some years. A form of chart that we have found especially valuable is shown herewith as Fig. 9. It will be noted that practically all the information which is needed by the executive to make decisions as to the desirability of changing the present manufacturing method are found on this chart, and since the number on the individual square of the charts refers back to the individual operation folder in the file, which contains all the supporting informations, it is easy by making use of such a chart, to arrive at a program for rearranging manufacturing operation where financing must be very seriously considered.

The article further refers to the use of micromotion study in studying the various operations contained in the process chart. The use of micromotion study in this way will be found especially useful, since the micromotion chart as made up represents the cycles of motion by different colored patterns. The similarity in the pattern points out immediately to the engineer which operations should be combined, and which operations should be sub-divided. We have been able to make special use of this feature of the micromotion studies in a number of different instances. For instance, one large electrical manufacturing company had a series of nine operations performed on one table. A micromotion study made of the nine operations showed that operations 1, 2, 3, 4, and 6 were of a similar pattern on the micromotion chart, while operations 5, 7, and 9 had entirely different patterns on the micromotion chart. The solution was very simple. We devised a machine which would enable one operator to combine all of the similar operations. The result is, that today, a blind girl is performing the five operations in the same time as formerly required three experts to perform the same five operations.

In a second case, a large corporation manufacturing special equipment, had 20 operators performing one assembly, whereas the finished operation called for two girls to do the work of 20. The difficulty with a great many assembly operations is that the attempt to specialize has been carried too far, in many cases, and not far enough in others. The wrong system of specialization has been used, and operators specialize on similar motions. The very natural

result is that there is a consequent loss of efficiency, a discontent on the part of the operator, and a backing up of materials in the plant, whereas, if the operation is divided according to the similarities of motions required, a single operator can become expert on a series of motions, automatic machinery can be installed in many cases, and the cost of production reduced to a point which looks like a day dream when first mentioned.

Mr. Gilbreth's process chart, which indexes the micromotion charts, and other supporting data, offers a backbone of what would appear to the final development of manufacturing economy.

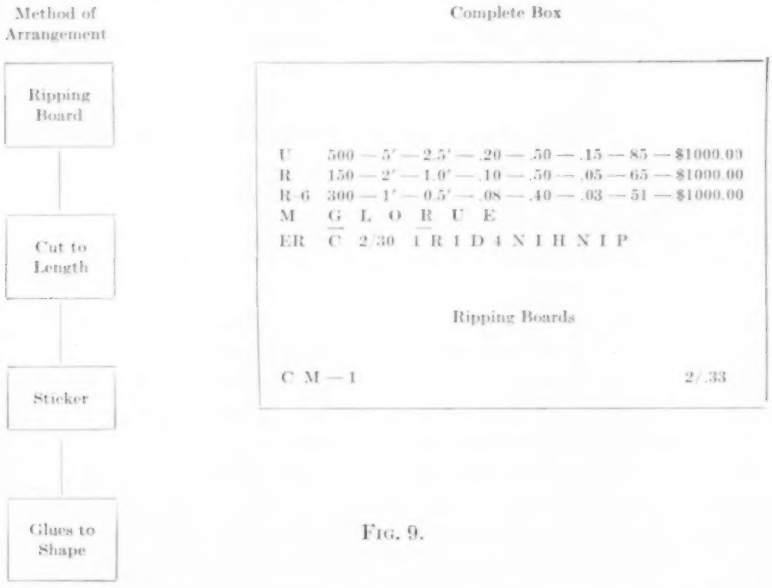


FIG. 9.

The following is a general form of routing chart, with an illustration made from a firm manufacturing table tops. The routing chart is arranged as shown on the left-hand side of this sheet, as shown in Fig. 9. A single box with all the information shown, is on the right-hand side of this sheet. The layout at the left-hand side of the sheet is self-explanatory, being the same general scheme as used by Mr. Gilbreth. The right-hand side of the sheet can be explained as follows:

S = Space

T = Time which one manufacturing unit would require to turn out a single piece of the material

Th = Time (T) divided by the total number of manufacturing units in the plant
L = Present labor costs per unit
M = Present material costs per unit
O = Present overhead cost per unit
T = Total cost per unit
E = Present equipment value.

The horizontal divisions are as follows:

U = Amount used at the present time on each of the items specified
R = Amount that would be required if all the equipment were brought to highest efficiency
R–6 = The amount that would be required if all units were brought up to highest efficiency, and six times the output were required
M = Motion
ER = Employment requirements.

Under "Motion" we have:

H= Search
S = Select
L = Transport loaded
O = Position
A = Assemble
B = Disassemble
F = Find
G= Grasp
E = Transport empty
U = Use
I = Inspect
C = Avoidable delay
N = Unavoidable delay
T = Delay for overcoming fatigue
P = Pre-position for next operation
R = Release load.

The heavy lines beneath the "G" and "R" indicate that the "Grasp" operation and the "Release Load" operation require more time than necessary and that changes on these machines should take the form of investigation as to how these two fundamental motions may be reduced.

Under "Employment Requirements" beginning at the left-hand side of symbol, the letters represent the following:

C = Intelligence army test standard
$\frac{2}{30}$ = The fraction of a month that it would require a new man to learn the operation
1 = Number of arms required; indicating that if a man has either arm, he can handle the job. It is not material which arm he has
R = Refers to his nerves, and indicates that his nerves should be reliable, in other words, the man should not have any disease which prevents the full use of the brain at any time
1 = Indicating that a man must have one hand
D = Indicating that the man may be deaf
4 = Indicating that the man must have four fingers
N = Representing that the four fingers must be nimble
1 = Representing that the man must have at least one leg
H = Representing that hernia is permissible on this particular job
N = Representing that a man's lungs must be normal, in other words that tuberculosis should not be permitted at this point
I = Representing that irritated skins so long as it is not due to a disease which is commutable, is permissible
P = Representing that first-class eyesight is not required but a man may be employed who must wear strong glasses.

The operation itself is ripping boards. In the lower left-hand corner will be found a symbol C M — 1 which means:

C = That this is part C referring to cleats on table tops
M = Operation is a machine operation
1 = That this is the first operation in the line-up of the cleats.

On the lower right-hand corner we have "2" which refers to the number of men required to operate this machine. "0.33" which indicates that these machines are running 33 per cent of their time, approximately.

THE AUTHOR agrees completely with what Mr. Clark has said, that the making of the process chart would make the foreman think more clearly on the problem before him. Mr. Clark said a very im-

portant thing when he said the mere recording of the fact will pay for itself. However, we tried to record these facts in the cheapest possible way and usually we make these charts in a very plain and inexpensive manner, but sufficiently in detail to enable the general process to be completely visualized.

Mr. Cooke has asked an important question — where is the place of the process chart in the sequence in relation to the route chart. It is a part of the definite scheme of visualizing *what has been* and *what is to be* done.

Mr. Taylor has quoted the definition that "management consists first, of finding out what you want to do, and then doing it in the best way." That might be worded more to the writers' liking if he had said — "first find out what they did, and then find out what they should do, to do it in the one best way, and then do it in that way."

In a complete scheme of visualization the process chart covers two steps; first, the purpose of finding out what is done, and second, what the different men think they should do, and consequently, in its last analysis, it results as follows. If you are going to visualize thoroughly a business, you might perhaps get a best general view of its buildings and general layout by flying over it in an airplane and take a view from the sky or a photograph of it. An isometric drawing, such as the typical insurance company drawing, serves the same purpose. The next step in visualizing is to prepare a route model. After the route model the next step is the process chart which records present methods. The process chart describes what they now do and it becomes the base from which they make other process charts of proposed methods. Any change that is recommended or what they propose to do is put on another blueprint, and the original is not disfigured.

After the process chart the next step is the standard detailed route chart. The process chart in no way supplants nor eliminates the route chart. The route chart should be used to show detailed operations. After the route chart, come the route sheets, instruction sheets, tool lists, stereoscopic photographs, micromotion studies and other detailed devices for visualization when and if required in the complete plans of visualization.

No. 1819

STEAM-CONDENSING PLANTS

By Paul A. Bancel, New York, N. Y.

Junior-Member of the Society

THE heart of a power plant lies in the boiler room, and the most important aspect of the condensing plant is in its relation to boiler-room costs. The choice of a condenser may be said to rest largely on its influence in reducing boiler-room charges. As in-

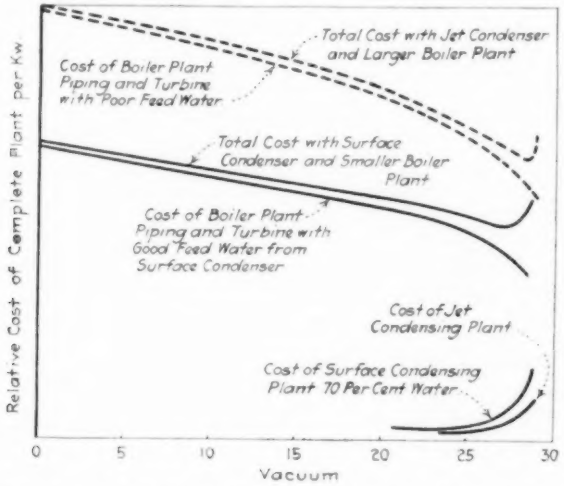


Fig 1. Relative First Cost of Steam Power Plant at Different Vacuums

dicated in Fig. 1, the decrease in cost of the boiler plant would make it profitable to run condensing if coal cost nothing; and pay even more, if the boiler feed was poor or expensive, to install the costliest of condensers — the surface type. With pure feedwater there is less fuel loss and boiler-maintenance charges, and, what is

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of still greater importance, boilers may be driven at higher ratings with consequent further reductions in first costsand fixed charges.

2 The designer of a power house is in a quandary — flying from boiler evils to condenser evils, and often with hesitancy as to which are the less serious. The problem is one of balancing fixed and operating charges in the boiler room against similar charges in the condensing plant.

FIXED CHARGES

3 The fixed charges against a surface-condensing plant depend on —

a Vacuum desired, which influences the size and cost of the equipment

b Cost of tube replacement, which depends on amount of tube surface in any particular design and life of the tubes

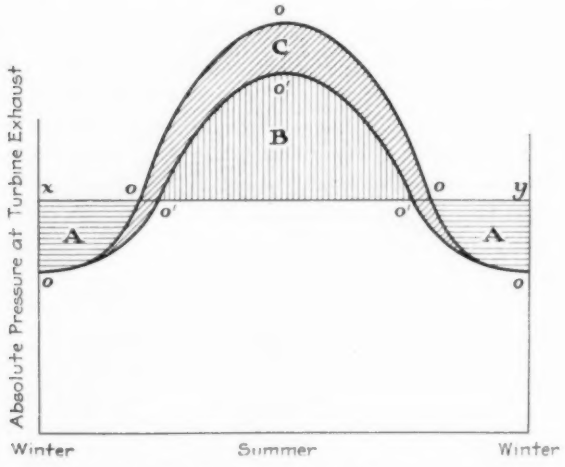


FIG. 2 CYCLE OF ABSOLUTE BACK PRESSURE OF YEAR

Line *x–y* represents highest vacuum or lowest desirable exhaust pressure. Areas *A*, *A* represent waste in winter due to excessive vacuum and areas *B* and *C* waste in summer due to warm water and insufficient vacuum. Area *C* indicates improvement possible by investin more money.

as influenced by water conditions and condenser design

c Power for the auxiliaries, which increases if the cost and fixed charges are reduced.

4 *Vacuum.* In Fig. 2 the curve 0-0-0 shows the absolute bock pressures at the turbine exhaust which might be expected from a cycle of water temperatures and a certain investment in conden-

ser. Frequently the design of the turbine is such and the water temperatures in the winter are so low that for certain winter periods a vacuum is obtained which is better than can be utilized by the turbine. On the other hand, the temperature of the available water supply for three to six months of the summer is 60 to 80 deg., so that the vacuums obtainable are lower than are desirable for maximum turbine economy. Area *C* in Fig. 2 indicates the range of vacuums by investing more or less money.

5 As will be seen later, extensive calculations in regard to best investment are apt to be misleading because of the reduction in vacuum which so often occurs due to fouling of the tubes. In such instances all other questions become subordinate to that of maintaining high vacuum continuously, under actual operating conditions. In other cases the decision must rest on the relation of load factor, months of the year having hot water and the fixed charges and operating charges at different vacuums.

6 *Cost of Tube Replacement.* This can be set up as a yearly depreciation reserve. The life of a condensing plant might be 20 years, whereas the tube life might be only five. On a 10,000-kw. turbine this might necessitate the purchase every five years of tubes costing about $10,000. Thus in determining the best vacuum the item of tube renewals may have a preponderating influence. Furthermore, as between two condensers for the same vacuum, one of which requires less tubes than the other, it is manifest that the fewer the tubes the less the fixed charges.

7 Similarly, the longer the tube life the smaller the fixed charges and cost of power. In one plant the tube life might be five years and in others a great deal more or a great deal less. Tube life depends on the character of the water supply, on the chemical, physical and metallographic properties of the tubes (crystalline structure) and it also depends on the design of the condenser. The latest report of the Corrosion Committee of the Institute of Metals, London, summarizes the types of condenser-tube corrosion, principally general thinning, pitting, apparent dezincification and water line attack at the inlet, due to local disturbances caused by bad entry. The fourth and fifth reports of this Corrosion Committee are exhaustive studies of the subject, and among other causes emphasize the effect on corrosion of low velocity and high temperature.[1]

8 Velocity has a direct effect on corrosion. Low velocity allows the deposit of foreign matter in the tubes which forms obstructions

[1] Jour. Inst. of Metals, vol. 23, p. 76.

around which the products of corrosion accumulate in concentrated form, with the resultant pitting. High water velocity keeps the tubes clean and eliminates this form of corrosion.

9 High temperature accelerates corrosion of all types and so seriously that a temperature of 95 deg. fahr. is recommended as the maximum. There are three ways of reducing temperatures in order to alleviate corrosion. by increasing the water quantity, thus decreasing the overall rise; second, by eliminating hot spots and local overheating of tubes; third, by increasing water velocity or decreasing tube size, thus decreasing the skin temperature at the tube wall.

10 In a condenser with proper distribution to avoid hot spots. and with suitable velocity and tube diameter to further reduce skin temperature and prevent foreign deposits, a markedly longer life of tubes will be obtained. As will be seen, too, these features of design result in small initial surface requirements, so that there is a twofold reduction in the fixed charges to take care of replacements.

OPERATING CHARGES

11 The principal operating charges on a condensing plant may be grouped as follows:

- *a* Reduction of condensate temperature below steam temperature and consequent loss of coal
- *b* Power for operating the auxiliaries
- *c* Reduction of vacuum and increased coal consumption due to scaling and sliming of the tubes
- *d* Cost and time lost to clean the tubes and cost and time lost due to tube failure and consequent pollution of the boiler feed.

12 *Reduction or Depression in Condensate Temperature* below steam temperature depends on condenser design and temperature of circulating water, being less with warm water and greater with cold water. The water of condensation must fall over successive rows of tubes to reach the bottom and if the steam is not drawn down to the bottom of the condenser, the water will be gradually cooled. The colder the water and higher the vacuum the greater the proportion of idle surface and the colder the condensate. Even in those condensers, in which numerous short-circuiting lanes are provided for the express purpose of securing penetration of the steam, there is a depression of condensate temperature of 5 to

10 deg. in summer and 10 to 20 deg. in winter. Eleven degrees loss of condensate temperature represents 1 per cent loss of efficiency and is equivalent to 300 kw. on a 30,000-kw. machine. In some plants this loss is largely overcome by the installation of preheaters.

13 *Power for Operating the Auxiliaries* has already been considered in a sense in the discussion of fixed charges, since the power required for any vacuum is partly dependent on the investment, and may be decreased by increasing the investment and fixed charges. No distinction should be made between the power required for driving condenser auxiliaries and the main power output of the plant, even where the auxiliaries are steam-driven and exhaust to a feed heater. The auxiliary power is a direct charge against the main unit output.

14 The principal item of power consumption lies in the circulating pump, and this unit is often arranged for variable-speed drive or else is divided into two smaller units, so that with cold water or light load the operating charge may be reduced by slowing down or by shutting down one pump. As shown in Fig. 2, an excessive vacuum is obtainable in the winter and the waste of fixed charges can be partially offset by saving on the circulating-pump power.

15 The best balance of circulating-pump power in the summer time for conditions of maximum water temperature and full load must be settled after consideration of all the aspects of the problem. By proportioning the condenser for low water velocities the head on the circulating pump may be made low, and the quantity of water used very low with small power requirement for driving the pump. This saving, however, may be counterbalanced by excessive fixed charges distributed over a relatively small part of the year.

16 Low pump power with low water velocities may be offset also by increased operating charges due to low vacuum resulting from fouling. Fig. 3 is similar to Fig. 2, except that a shaded portion *D* has been added to indicate the range of vacuum in the summer, time under actual conditions of fouling of condenser tubes. As much as an inch of vacuum is lost in a week in some cases, and in one extreme instance there is a loss of an inch in two days. It is unfortunate that the loss by fouling is most severe in the summer when it is most difficult to obtain high vacuum. As a reduction of only $\frac{1}{4}$ in. in vacuum represents a loss of $1\frac{1}{2}$ to 2 per cent of turbine economy, which may be equivalent to all the auxiliary power, it is readily seen how easily all calculations of exact balance between fixed charges against the condenser and operating charges against the auxiliaries may be disrupted.

17 *Reduction in Vacuum Due to Tube Fouling.* The foreign matter that accumulates in condenser tubes may be divided into three classes:

a Mechanical deposits, silt and mud
b Scales deposited out of solution
c Slimes or organic growths.

18 The first of these is of minor influence in reducing vacuum unless water velocities are low indeed. The ability of a stream of

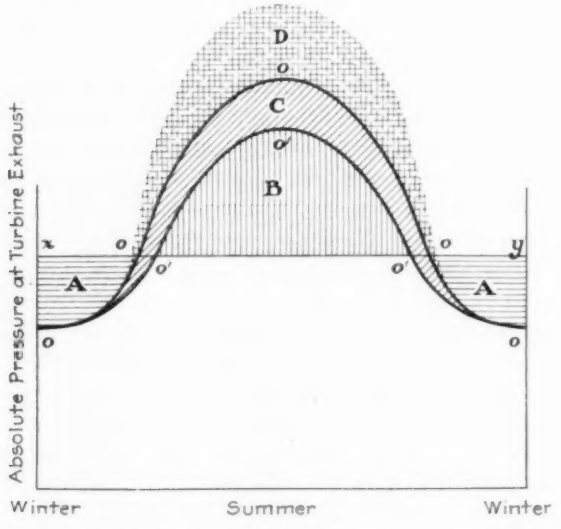


FIG. 3 CYCLE OF ABSOLUTE BACK PRESSURES AT DIFFERENT SEASONS
Area *D* shows increased back pressure or loss of vacuum due to fouling.

water to carry solid substances in suspension increases rapidly with increase of the velocity. However, foreign matter, such as silt, may be deposited in slime coatings which form a more adherent surface than a clean condenser tube.

19 The second form of deposits is common with salt water and hard waters carrying scale-forming salts, familiar in impure boiler feedwater. The commonest ingredients of such water are calcium and magnesium carbonate which are soluble in the presence of carbon dioxide. Heating the water drives off the CO_2 and precipitates the carbonates. It is probable that high velocity has a

mechanical effect in reducing the rapidity of such scale formations as well as reducing temperature at wall.

20 Where extremely hard waters are encountered it is desirable not only to keep down the temperatures by observing the principles of high skin velocity and acitivity but also by using larger quantities of water and smaller overall temperature rise. In this way it is possible as between two condensing plants to reduce the maximum temperature rise of the skin of water in the hottest tubes, 10 or 20 deg., or even more.

21 Slimes or organic growths are the third form of deposits. Fouling of this character is most severe in the summer time, and is also most severe in hot portions of a condenser with improper steam distribution. Thus, in a condenser with lanes, the tubes in the lanes will be found coated with a thick, heavy slime in a short time, whereas tubes in the short-circuited regions will be but slightly fouled. It is probable that these slimes are organic growths occurring under the ideal conditions of warmth and stagnation existing at the inner skin of the tube. A typical sample of condenser-tube slime was taken a short time ago by the author and given to a bacteriologist for analysis. A large number of hyphomycetes were demonstrated. This form of life is classified between bacteria and plants, as shown in Fig. 4. The hyphomycetes grow rapidly as long microscopic threads (Mycelia), which branch rapidly, forming a dense interlacing network in which all other organic and inorganic elements are caught. They grow better in an acid than an alkaline medium. The hyphomycetes are familiar to us as molds and fungi and are a form of life just below the algæ, which are the green slimes frequently seen in stagnant pools. The algæ contain chlorophyl and require light for their growth, whereas the hyphomycetes grow in the dark. The conditions for rapid growth are moisture, stagnation and a fair warmth. The optimum temperature is between 20 and 40 deg. cent. (68–104 deg. fahr.); when spores are formed they are more resistant to higher temperatures.

22 The same arrangement of tubes to prevent local overheating, and the use of high water velocities to reduce inner skin temperatures and increase the activity and counteract stagnation along the inner wall of the tube, may be expected to reduce and eliminate the formation of organic slimes. Corrosion, scaling and sliming can all be reduced by correct proportion of a condenser for high activity of water flow and turbulence at the tube wall and

uniform steam distribution, factors which in addition permit of a large reduction in the amount of tube surface.

23 Fouling adds to the operating charges in another way, as there may be a large labor cost for cleaning. The time loss with a unit out of service is also a considerable item of cost. The charges due to failure of tubes by splitting and corrosion are of a similar nature. The pollution of the condensate and consequent scaling of the boilers from either impure condensate or from raw feedwater

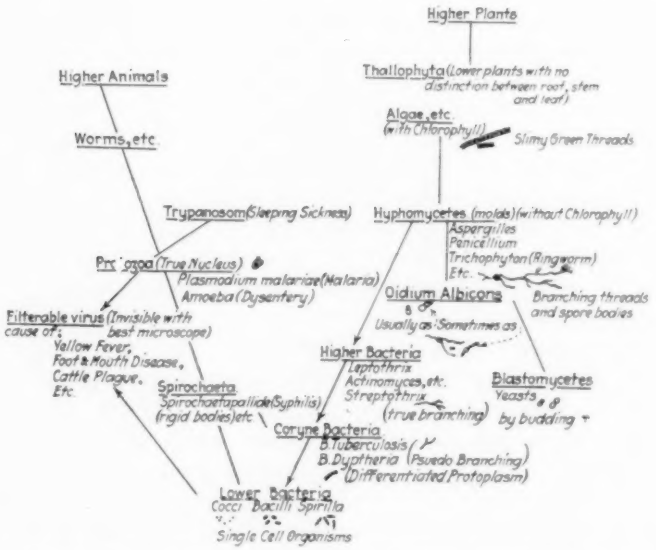


FIG. 4 VARIOUS FORMS OF VEGETABLE AND ANIMAL LIFE SHOWING RELATION OF SLIMES (HYPHOMYCETES) TO ALGAE, ETC.

adds to the operating charge in the boiler room. The loss of time on the complete power-generating unit required to shut down, locate, and plug the leaking tube adds to the turbine-room operating charges.

24 *Clogging of Tubes.* One other loss adding to operating charge is that caused by clogging of tubes by accumulation of miscellaneous solid substances against the tube sheet. This applies only to plants using natural water supplies. With spray ponds the nozzles clog long before the condenser tubes. Clogging can be reduced by proper screening, and the use of flush, well-rounded tube inlets and high water velocities. Low velocities are wanted in the screens, but high velocities and clear inlets into the tubes.

The influence of velocity is clearly shown by the tendency in two-pass condensers for the tubes of the return pass to clog with solid matter which originally came through the first pass. This clogging is most noticeable toward the top (where the water is needed most) and is due no doubt to the slow velocities at the top of the return box, which is at the top of the siphon and acts as a skimming tank.

CONDENSER DESIGN

25 The design of a surface condenser to conform with the requirements which have been outlined, lies in the proper application of certain well-known principles. The first of these is to keep all of the tubes uniformly hot on the steam side so that the minimum number will be needed because none is wasted. To accomplish this the steam must be free to flow to all the tubes and over all the tubes, with slight loss of pressure and temperature, the condenser being arranged in stages, much as is a turbine, with large flow areas at the exhaust entrance and great distances between rows to avoid sharp bends and pressure loss. Fig. 5 illustrates the analogy.

26 Of equal importance in keeping all tubes hot is the proper staging to maintain velocities that will sweep the air ahead in the current of steam and the arrangement of tubes in staggered rows so that none is shielded or pocketed. Air left on a tube after condensation would otherwise stratify in a thick layer in a few seconds. With rapid steam velocities as shown in Fig. 6, all of the tube is hot because the air is continually carried away in the current of steam. Experiment has shown that this current is most active in producing heat transfer in front of the tube; the side about 90 per cent active, and the back about 75 per cent. This indicates the importance of a staggered tube arrangement so that the front of each tube is presented to the steam flow.

27 The steam velocities must be sustained throughout the depth of the condenser, which not only requires proper tube spacing but a shell of decreasing width — that is, a wedge- or heart-shaped shell terminating in a narrow, slot-shaped outlet. In this respect also the condenser is analogous to the turbine, Fig. 5, which has both changing blade height and changing wheel diameters.[1]

28 If the foregoing conditions are maintained, the mixture of

[1] Wedge-shaped shells have been in use for many years with Weir's marine condenser. The earliest record of this type of shell known to the author is that of its use 33 years ago by another English manufacturer.

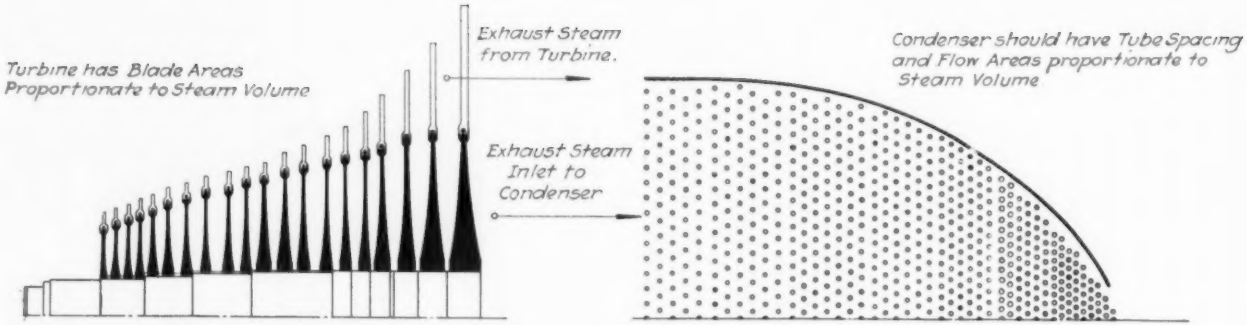


FIG. 5 COMPARISON OF STEAM FLOW IN TURBINE AND CORRECTLY DESIGNED CONDENSER

steam and air is kept homogeneous and the temperature does not fall until a surprisingly large percentage of steam has been abstracted and the process almost completed. Fig. 7 shows the relation between percentage of heat removed from an average steam-air mixture such as would enter a condenser, and the temperature of the mixture. Starting with approximately 92 degrees temperature

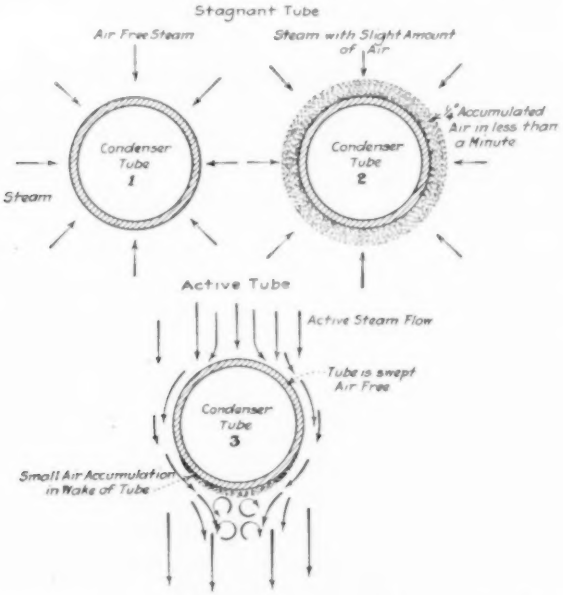


FIG. 6 IMPORTANCE OF STEAM VELOCITY IN PREVENTING STRATIFICATION OF AIR AROUND A CONDENSER TUBE

Stagnant Tube: Condition when steam flow is produced by avidity of tube itself for condensing steam.

Active Tube: Condition when there is active steam flow to sweep tube free of air.

there is no perceptible fall of temperature due to increasing air richness until over 99 per cent of the heat has been abstracted and over 99 per cent of the steam condensed. Thereafter the temperature falls with a marked knee in the curve, the exact position and shape of which depend on the proportion of air to steam in the original mixture.

29 This process is *no longer rapid condensation but a matter of devaporization*, to reduce vapor temperature and pressure and increase partial air pressure so that the air may be removed by a

vacuum pump. It is best carried out in a chamber external to the main condenser forming a devaporizer or cooler. Figs. 8, 9, and 10 illustrate such an arrangement in which the flow of hot vapors and air taken from the bottom of the main condensing chamber is turned upward into a cooler from which the concentrated air is finally removed at the top. It will be noticed that this cooler is also of wedge shape, decreasing in area as the outlet

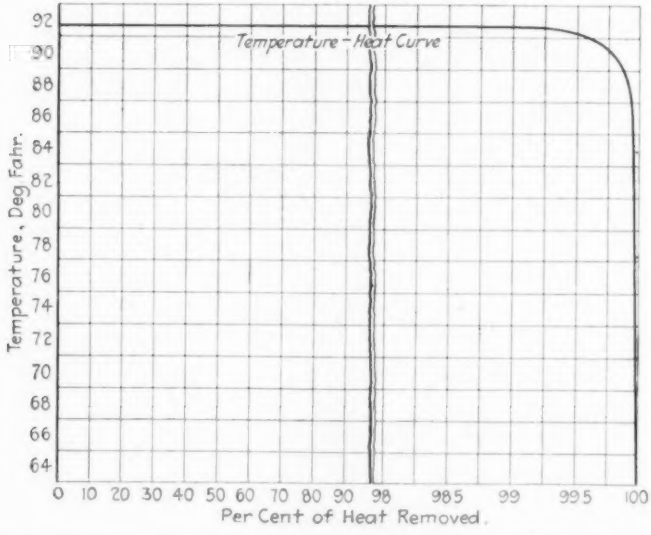


FIG. 7 RELATION BETWEEN PERCENTAGE OF HEAT REMOVED FROM A MIXTURE OF STEAM AND AIR AND TEMPERATURE OF THE MIXTURE

Note scale is magnified 40 times at right-hand section of chart for the assumed mixture which is a fair average in high-vacuum condensers; the temperature starts to fall at 99 per cent.

is approached. The air is withdrawn by steam-jet augmenters which discharge into an intercondenser section forming part of the cooler. The air at reduced vacuum is then withdrawn and compressed to atmosphere by secondary steam jets or a small reciprocating pump. In one case a single small pump is used as a secondary for two condensers, steam-jet secondaries being installed as stand-bys.

30 The work of devaporization is done by cast-iron grids with fins, meshed together as illustrated in Fig. 11. Cold circulating water flows through the cores of the grids with only slight rise in tem-

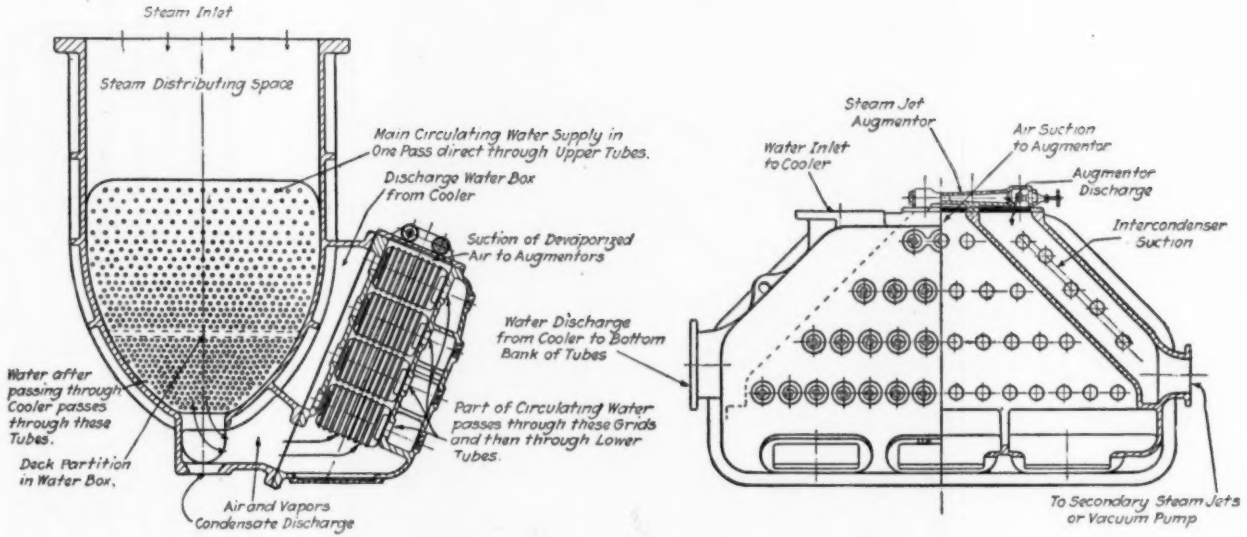


FIG. 8 CROSS-SECTION THROUGH MAIN CONDENSER AND COOLER OF INGERSOLL-RAND SURFACE CONDENSER

perature and then passes through the bottom group of tubes in the condenser (Fig. 8).

31 A large experimental plant, for capacities up to 70,000 lb. per hour was built at the Phillipsburg works of the Ingersoll-Rand Company to investigate these departures in design. The photograph reproduced in Fig. 9 illustrates the complete plant. By the use of electrical thermometers the performance of each row of tubes was determined and from that the efficiency of the tubes

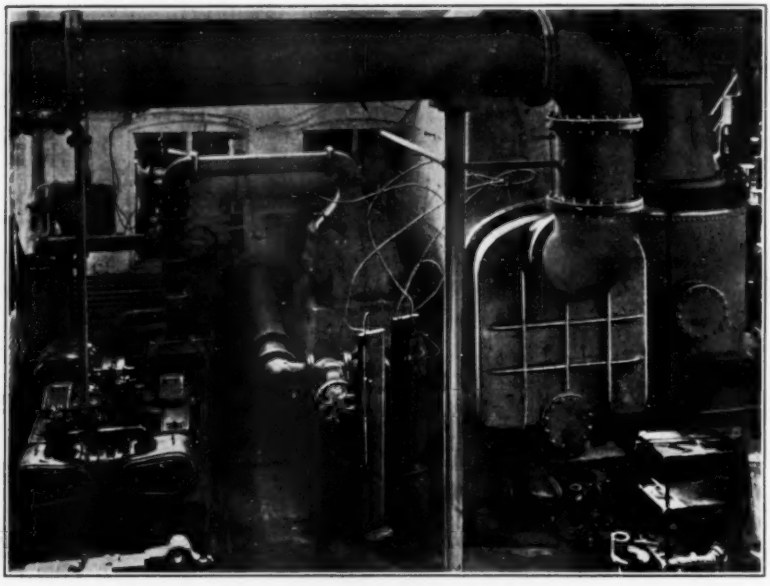


FIG. 9 LARGE EXPERIMENTAL SURFACE-CONDENSER PLANT AT WORKS OF THE INGERSOLL-RAND CO., PHILLIPSBURG, N. J.

under all conditions of loads, vacuum and air leakage and in different parts of the condenser. A sliding or "trombone" thermometer of the electrical resistance type in the discharge water box was arranged so that it could be shifted to any position desired. Decks were fitted in the box which could be shifted to vary the water distribution so that a wide range of conditions of water circulation was obtained. It was found that high efficiency was obtainen down to the last row of tubes of the condenser so long as certain critical conditions of vapor velocity were maintained. The con-

denser was hot throughout its depth and the condensate water at practicallp steam temperature. The hydrodynamic loss or pressure drop was also studied by the use of water manometers, from which the important relations of pressure loss, steam velocities, tube spacings, and number of rows were determined.

32 Depending on the air leakage, the cooler devaporizes and

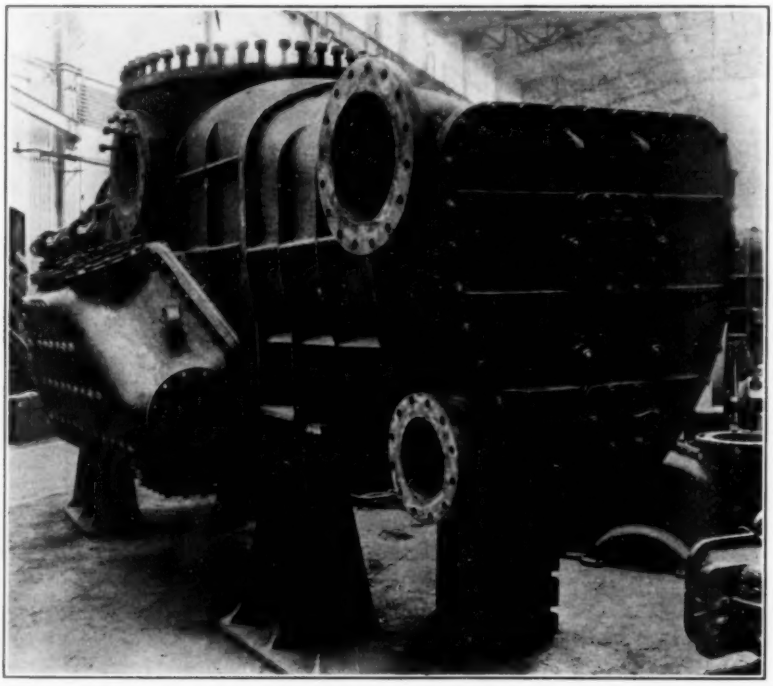


FIG. 10 INGERSOLL-RAND SURFACE CONDENSER FROM INLET WATER BOX-END

At the left is the cooler, mounted on shell.

cools the air down as close to the temperature of the cold circulating water as one degree, thus giving complete countercurrent action. A study of these temperatures was made by electrical thermometers located at successive points as shown. As is well known from Dalton's law, the amount of air contained in a homogeneous flowing mixture of steam and air depends on the partial air pressure, which in turn depends on the temperature and corresponding partial

vapor pressure, so that complete countercurrent action gives maximum withdrawal capacity of the vacuum pump and protection against excessive reduction of vacuum due to heavy air leakage.

33 It is evident that with the cooler taking care of the process at the knee of the curve and the main condenser hot and active throughout its depth, water cannot pass through great groups of

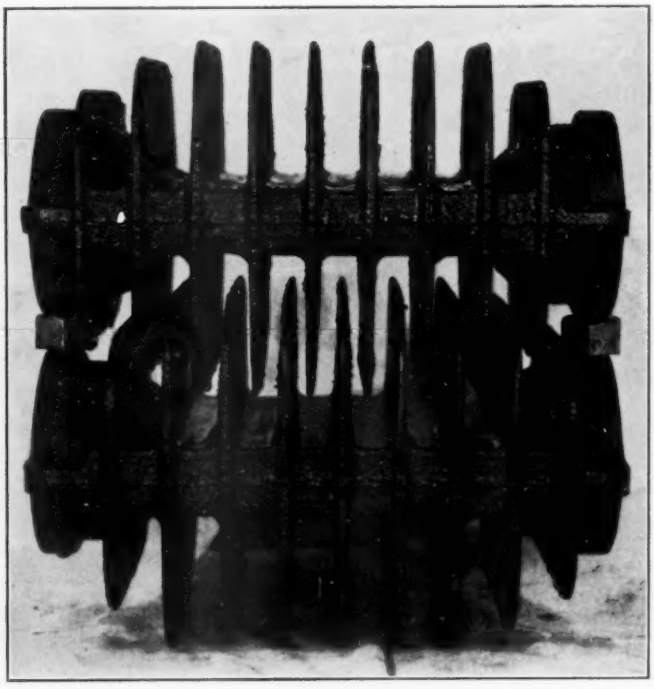


FIG. 11 SHOWING GRIDS AS MESHED WHEN ASSEMBLED IN CONDENSER COOLER

tubes without heating, and therefore the entire bulk of the circulating water can be heated with only one passage through the tubes instead of multiple passes. The circulating water may be pumped, therefore, at low head and power consumption, with high average rate of condensation since none of the tubes is wasted, or else at greatly increased velocities and correspondingly increased rates of condensation. The main supply of water passes directly through the upper and larger group of tubes and the remainder of the water

after passing through the cooler, goes through the lower and smaller group of tubes below the deck or partition.

34 The rate of condensation in a condenser with all its tubes active was found to vary substantially as the rate of condensation on a single tube, which, as is well known, depends on its diameter and material and the water velocity. With only one passage necessary, an entirely new basis for selection of velocity and tube size is opened

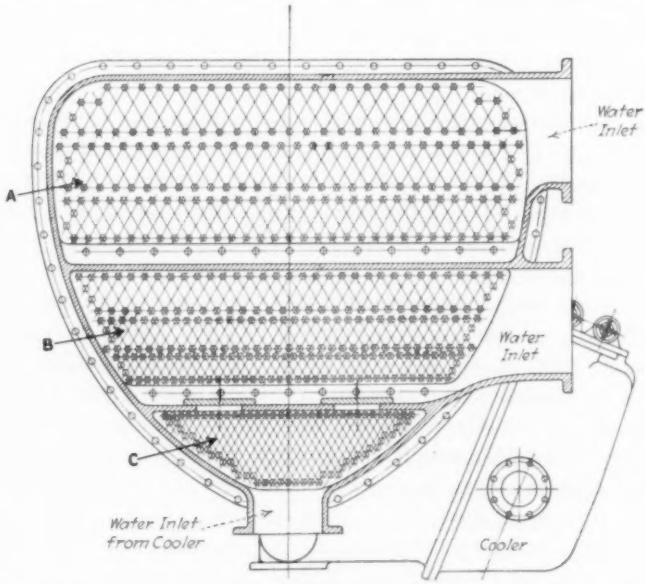


FIG. 12 WATER BOX WITH TWO INJECTION NOZZLES ON I.-R. SURFACE CONDENSER

Both nozzles are used in summer, the upper one only in winter, thus obtaining high water velocity even with reduced water quantities and concentrating condensing action in top of condenser.

to the designer. High velocity or small tubes or both can be used without excessive pump power as would be encountered with multiple-pass construction. Paradoxically, there is more insurance against corrosion, as well as greater protection against reduction. of vacuum by scaling and sliming, when working the condenser at the higher and apparently less conservative condensation rates. The author does not wish to imply that excessive pump powers should be used. High capacities and velocities may be obtained without exceeding circulating-pump powers now often used. The circulating-

pump capacity is best arranged in twin units so that maximum water quantity and velocity are available in summer and can be reduced in winter by shutting down one pump. Variable or graded circulating-pump capacity is desirable in small as well as large units.

35 On large units having two water pumps, one of which is shut down in winter, provision can be made for preserving high velocities even with reduced water quantity. At high vacuum with cold water, the top tubes in large condensers do most of the work and therefore the smaller winter quantity should be concentrated on these tubes, thvs maintaining high velocities in the tubes which tend to foul and corrode.

36 In the summer time both pumps supply water to the two nozzles (Fig. 12) serving sections *A* and *B*, and a small quantity of water is supplied the coolers and from them to section *C* at the bottom of the condenser. In winter one pump is shut down, the valve to section *B* shut, and the plates in the deck between *B* and *C* removed thus throwing these two tube banks in parallel so that they both receive water from the coolers. The main water supply is then concentrated on the upper section *A*. This design also has the advantage that at any time the entire capacity of the two pumps may be concentrated on section *A* or *B*, thereby producing extremely high cleaning velocities, also, that either section *A* or *B* can be shut down independently and the water-box covers removed for inspection or cleaning.

DISCUSSION

P. E. Reynolds. The heart- or pear-shape condenser shell was used possibly thirty years ago with excellent results. The steam-jet type of air pump was used over twenty years ago and yet the idea has been taken up and developed only in the past few years. The development of surface-condenser design has certainly wandered in what might be called a meaningless and aimless manner in the past twenty or thirty years. Starting with the old Weir pear-shape condenser, which we have now rediscovered was a really meritorious and theoretically correct design, we seem to have turned, as larger units were required, to a rectangular or cylindrical shape of shell, this shell being packed with as many tubes as possible. It was next decided, apparently, that it might be better to take out some of the tubes, leaving more steam space and allowing easier access of steam to the remaining tubes. We now appear to be headed back to the original

design, having a pear-shape shell, or its equivalent, with graduated tube spacing, the steam passages varying in proportion to the volume of the steam passing.

The writer is inclined to believe that the seemingly aimless course of change or improvement in condenser design has been due to the difficulties and expense involved in making condenser tests and in altering existing designs. Condensers may easily be tested when installed in power stations in connection with the turbines or engines for which they were intended, but these tests give only negative information as to what may be expected of radical change in design or shape of condenser shell, graduated tube spacing, change in the path of circulating water, water velocities, etc.

Numerous and extensive tests have been conducted showing the heat transfer which may be obtained with a single or a small number of condenser tubes in an atmosphere of practically pure steam and the variation of heat transfer with water velocity is well known. It is also a very well known fact that the heat transfer obtainable with present condensers in actual operation nowhere near approaches that which can be obtained with a single tube.

The author's paper leads us to believe that he has developed a design with which such results are assured. It is regrettable, however, that no figures or results are given to assist us in arriving at conclusions. It would seem that a real improvement in condenser design has been accomplished. Graduated tube spacing, giving uniform steam flow and accessibility to cooling surface, should certainly tend to improve heat transfer. It would also seem very logical that the pear-shape condenser shell should be best adapted for most effective collection, and efficient removal, of air. The great question involved, it seems to the writer, is the extent of the improvement made in heat transfer and the resultant reduction in the tube surface required, and the balancing of these factors against possible difficulty in increased cost of manufacturing. In other words the problem is commercial; is it a paying proposition to utilize to the fullest extent the improvements in condenser efficiency which may be obtained by adopting a purely theoretically correct design?

There are a number of condensers on the market which use a materially wider tube spacing in the top pass of the condenser than in the lower pass. Other condensers use steam lanes or paths through the tubes to carry steam to the lower and less accessible tubes. Neither of these methods of steam distribution has the theoretically correct feature of the graduated spacing but nevertheless it has been

demonstrated by numerous tests on the largest units that they are effective. On many tests of large two-pass condensing equipments it has been found that the temperature rise of the circulating water through the lower pass was equal or slightly greater than that through the top pass. This clearly demonstrates that steam is being distributed over the lower as well as the upper tubes and that the lower tubes are doing their share of condensation. Other conditions which point to the same conclusion are the facts that with these designs the condensate temperature is close to the vacuum temperature and the pressure drop or friction through the steam space is slight.

The fact that heat transfer in a large number of condensers of good design is low is due largely to the excess surface which is usually furnished as a factor of safety to take care of fouling of the tube surface. It is unfortunate as the author states that the greatest fouling generally occurs in the summer when the circulating water is the warmest and when the largest amount of cooling surface is required. These are factors which both manufacturers and users have found by sad experience make it expedient to install a larger surface than would otherwise be required.

Most condensers of recent design contain an internal air cooler or devaporizer made by baffling off a small portion of the tube surface. This allows the air vapor passing to the air pump to be taken from the coolest surface and at the lowest temperature. Although it would seem to be a difficult matter to determine just where condensation ceases and devaporization or cooling takes place, as these terms are entirely relative in their meaning.

The author's design, using only one pass of the circulating water through the condenser, has many advantages, provided it does not entail too long a tube length in order to maintain a suitable water velocity. Assuming the usual 70-deg. fahr. circulating water and a 15-deg. rise with a heat transfer of approximately 400 B.t.u. the tube length for a single-pass condenser would figure out around 40 ft. to maintain a 6-ft. water velocity. Of course, if double the heat transfer or more is assumed or if the quantity of circulating water is doubled, the tube length could be reduced to 18 or 20 ft., which might be permissible.

Doubling the heat transfer however would probably not leave much of a factor of safety, if any, to cover dirty tubes, and doubling the quantity of circulating water would probably lead to an undesirable power requirement for the circulating pump.

The problem is one which the author has no doubt given close study and its solution might be more apparent if we had his test figures available. On the whole the writer believes that the author's ideas are leading in the right direction and that a material improvement in condenser design has been called to our attention.

A. G. CHRISTIE. The author's paper on condensers is the most interesting one on this subject that has been offered to the Society in recent years. He has presented a number of new ideas on condenser design and operation that deserve careful thought and study by all power engineers. Many sections of the paper could be discussed much more fully with profit. For instance, many engineers would welcome a more complete discussion of the causes and nature of tube corrosion than is possible in a paper of this limited length. A better understanding of corrosion would result in the taking of more intelligent steps to minimize its effects. The author has suggested some causes, the clear understanding of which should certainly lead to better results in handling condensers.

The arguments presented in this paper and the deductions therefrom lead logically to the heart-shaped condenser that is shown. As the author points out, there is no novelty in this particular form. The air cooler or dehumidifier is however of novel form and design. Engineers will be interested in some general data on the performance of this new condenser, such as the following:

1 High water velocities through the tubes are inferred. What are the desirable water velocities for best economical performance?

2 What is the friction pressure loss through this single-pass condenser with the high water velocity?

3 High water velocity in the tubes and rapid removal of air from the outside of the tubes should result in a greater rate of heat transfer through the tube surface. Have such greater rates of heat transfer been obtained?

4 Would it be advisable to increase the terminal difference between the steam and the leaving water in older condensers securing cooler tube conditions in order to reduce corrosion?

It is interesting to note that the condensate and steam in the condenser were at practically the same temperature in the tests at Phillipsburgh. These tests were undoubtedly made with full

load on the condenser. It is logical to expect a certain difference in temperature with less load.

EDWIN B. RICKETTS. The author has afforded us a very interesting insight into his condenser design, as far as it goes. As we all know, the heat transfer in a single-tube condenser is considerably over 1,000 B.t.u., probably one-third to one-fifth of that amount in the ordinary condenser.

We can undoubtedly approach the heat transfer coefficient of a single-tube condenser by enlarging the spaces around the tubes. It is a matter on which we must have figures, however, to tell whether or not we are going to save money by increasing these spaces. It is well known that by increasing the velocity of circulating water we can increase the heat transfer coefficient but this increase in velocity may be bought at a power requirement cost which would make it very undesirable.

Probably the author has determined the heat transfer coefficient which he obtained in the different passes of his condenser, and it is hoped that he will give this information, so that it will be possible to form a better idea of this condenser in comparison with others. It would also be interesting if the author would give some data on the air cooler of his condenser.

The author brought up a very interesting bacteriological exhibit in discussing the slimes in condensers. Some years ago the writer had a bacteriological examination made of the slime in the condensers using water from the East River and the report showed that this water was almost absolutely sterile, due to the effect of copper salts. This examination was made with the idea that possibly the health of our men would be affected by handling this condenser slime, but the report showed that the slime was absolutely harmless.

D. K. DEAN. The writer is always interested in reading papers presented by Mr. Bancel as these papers have quite generally included many novel and instructive ideas. In the present paper, the writer finds several points to which he can hardly subscribe.

The writer agrees fully with the author in the desirability of maintaining a high water velocity through the tubes of a condenser, placing the limitations, however, at the point at which the cost of power to circulate this water offsets the compensating gain, whether in improved vacuum or reduced maintenance charges.

The writer does disagree, however, with the statement that

"no distinction should be made between the power required for driving condenser auxiliaries and the main output of the plant even where the auxiliaries are steam driven and exhaust to a feedwater heater."

This statement is tantamount to the assertion that there is no difference in the thermal efficiencies of steam as opposed to electric drive. This is rather outside of the facts, since an auxiliary driven by current received from the main unit would have the switchboard steam rate of the main unit, which would reduce to say 10 to 14 lb. per brake horsepower hour plus its share of the entire overhead of the installation, while the charge against a steam driven unit exhausting to a feedwater heater in which a heat balance is maintained would be in the neighborhood of but $2\frac{1}{2}$ to 3 lb. per brake horsepower hour plus this unit's own overhead, and its share of the overhead of the boiler, piping and return systems only. This statement the writer believes needs limitations.

Considering the paper as a whole the logic of the author's argument is to prove the superiority of the pear section condenser, which he describes, as opposed to condensers having more orthodox methods of tube spacing which rely upon steam lanes for leading the steam to the inner parts of the tube nest. The writer has no especial criticism to make of the pear-shape tube layout. As the paper states, this type has been used in England for years with good results as far as vacuum is concerned. It is to be recalled, however, that until quite lately English practice made use of the wet-vacuum system in which especial attention must be paid to cooling the condensate several degrees below the vacuum temperature in order to secure a reasonable volumetric efficiency of the wet-vacuum pump. A pear-shape layout with its diminishing cross-section requires the condensation to fall through the inverted apex of the section at which point the partial air pressure is highest with a resultant low transmission rate from the vapors to the cooling surface. The falling condensate under these circumstances tends to take the temperature of the tube rather than retain the temperature of the vacuum.

Of course the author advocates the use of a separate air cooler to reduce the volumes of the vapors. This is a distinct improvement over the English wet-vacuum practice. It is evident that the surface within the condenser should be proportioned so as to condense all the steam up to the "knee af the bend" in the temperature curve under the full-load summer water temperature condition,

as otherwise live steam would find its way to the vacuum pump suction. The requirement of meeting this extreme condition rather works against the apparatus at light load or with colder water temperatures, since in this case with the full amount of surface working the live steam would not reach the lower portion of the condenser and because of the depth of the air-bound section resulting from the shape of the shell, it would be expected that the condensate would be cooled several degrees below the vacuum temperature.

It is quite possible that this is one of the reasons why the athruo advocates the rather complicated method of water distribution in which an extremely sluggish flow would be secured in the lower banks of the tubes in the colder months by passing an extremely reduced quantity of water through these tubes. The result would be that the steam would possibly reach the bottom of the condenser and probably the temperature of the condensate would be reasonably near the temperature of the vacuum, a result secured, by the way, without complication with the lane type of steam distribution which the author decries.

Generally speaking it appears to the writer that the author's statements regarding tube layout in which steam lanes are used to assist steam distribution are a little strong without supporting evidence. Shop tests and also field tests of condensers with lane steam distribution indicate that with a proper design of this type, the work is quite uniformly distributed through the whole condenser. With a suitable number of lanes properly arranged there is no reason to assume that the tubes in the various banks between the lanes would be short circuited, since even allowing for the decreased resistance to flow within the lanes, the resulting velocities within the various banks of tubes will still be sufficiently high to sweep away air accumulations in that portion of the condenser in which the steam lanes are located.

As a matter of fact, there is no difficulty in providing a tube layout, in which all portions of the cooling surface in a given transverse cross-section are required to do their fair share of the work of condensation. The principal obstacle in approximating perfect distribution is that the usual physical limitations are such that it is difficult to distribute the steam equably to the ends of the condenser. The design advocated by the author is markedly deficient in this respect.

The author advocates a single-pass condenser. We may assume the following set of conditions and compare the effects upon steam distribution in a single-pass and in a two-pass condenser.

Suppose that a vacuum corresponding to 1.5 in. absolute pressure mercury is being maintained with water entering at 70 deg. fahr. and leaving at 85 deg. fahr.

In a single-pass condenser, under these conditions, assuming air-free steam, the water temperature distribution along the tubes would be approximately as shown in Fig. 13.

It will be apparent that on the inlet side of the center line, the rise in temperature will be

$$79.7 - 70 = 9.7 \text{ deg.}$$

while on the outlet side of the center line the rise will be only

$$85 - 79.7 = 5.3 \text{ deg.}$$

Since the amounts of steam distributed to the two sides of the

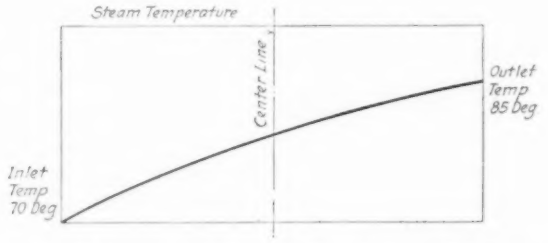


FIG. 13 TEMPERATURE RISE IN SINGLE-PASS CONDENSER

center line will be proportional to these temperature differences, it will be apparent that the amount condensed by the outlet half of the condenser will be only $\frac{5.3}{9.7} = 54\frac{1}{2}$ per cent of the amount condensed by the inlet half of the apparatus, a very poor distribution of work indeed.

With a two-pass condenser having equal amounts of surface in the two passes, the temperature distribution would be about as shown in Fig. 14.

Rise first half of first pass	$= 75.5 - 70$	$= 5.5$
Rise second half of last pass	$= 85 - 82.7$	$= 2.3$
Total rise left side of center line		7.8 deg.
Rise second half of first pass	$= 79.7 - 75.5$	$= 4.2$
Rise first half of second pass	$= 82.7 - 79.7$	$= 3.0$
Total rise right half of center line		7.2 deg.

Ratio of amount of steam condensed on right side of center line to the amount condensed on the left side of the center line $\frac{7.2}{7.8} = 92.3$ per cent.

This indicates a very fair division of work longitudinally and it is greatly superior to that which would be obtained with the design suggested by the author.

This comparison indicates the potentialities of different sections of the surface within a single-pass condenser and within a two-pass condenser.

Under actual conditions, in order that the sections of surface on the two sides of the center line should receive the amounts to which their potentialities entitle them in the case of a single-pass condenser, it will be quite obvious that there would be a very con-

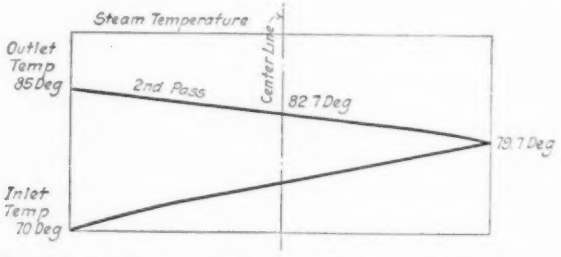


FIG. 14 TEMPERATURE RISE IN TWO-PASS CONDENSER

siderable whirl and change in direction of the flow of the steam in order that one end of the condenser may condense practically twice as much as the other end. Such a whirl of steam would without question result in an air pocketing of a fair amount of surface within the condenser with a resultant loss in the effectiveness of such surface.

Actually the steam distribution will be determined by the frictional resistance to flow from the steam inlet to the air outlet which will mean that it would be impossible for practically twice as much steam to be crowded through one end of the condenser as through the other end. The result would be that an appreciable amount of the condensing potentialities of the surface could not be utilized with the single-pass construction while with the two-pass construction the division of the work is practically equal on the two sides of the center line of the apparatus.

It is rather difficult to understand why the author emphasizes

the use of extreme care in determining proper transverse cross-sections in condenser design while at the same time he recommends a design which would give such a poor longitudinal distribution as above indicated.

As a matter of fact, the writer is somewhat confused by the argument set forth in the paper in favor of a single-pass condenser.

When laying out a condenser, the designer is presented with the fact that he is tied down to the use of commercial sizes of tubes which experience has indicated are most suitable for condenser work; also he is faced with a certain ratio of cooling surface to the amount of steam condensed as determined by the exact conditions of the installation. This ratio, to meet a given set of conditions, is well established and must provide a liberal margin of safety to cover periods of operation when the condenser may be fouled or during which there may be excessive air leaks. The proportions set by good practice in this respect invariably drive one to the use of two passes in the condenser in order to use commercial sizes of tubes and still get high water velocities.

It is this fact that makes the advocacy of the single-pass condenser rather confusing to the writer. It would seem that in order to use a single-pass condenser and still get high water velocities, it is the author's intention to reduce the amount of cooling surface to meet a given condition to a point much below that which experience has established as being good practice.

It would seem to be unwarranted to follow out such an intention in view of the deficiencies in the design as outlined above.

The writer can see no reason for departing from established practice in the proportioning of condensers.

JOHN F. GRACE. The author appears to assume that proper reduction of flow-area for maintaining correct velocity of steam through a surface condenser necessitates a heart-shape shell. The reduction he desires is obtained in the stronger cylindrical shell in the Morison (British) design in which longitudinal plates divide the shell into wedge-shaped condensing sections in parallel; in the Westinghouse which employs radial inward flow to a central air offtake; and in the Worthington, in which the double plates of bottle-tight coolers guide the steam flow through a reducing channel, and serve also to drain the condensate to the hotwell without permitting it to flow over the cold lower tubes.

We prefer lanes to successive reductions of pitch as shown

by Fig. 12, which were used by Worthington as early as 1904. We have not experienced severe fouling of tubes adjoining lanes.

The author favors single-pass in order to use small tubes and somewhat increase water velocity, apparently ignoring the glue-like, screen-defying liquids many of us must circulate. In New York power stations, one-inch tubes are now the rule.

A speed of about 30 feet a second will remove slime, but the pumping power for such tube velocity is out of the question. It is economy to use moderate velocity and to figure upon occasionally cleaning tubes. Rubber plugs or Bullett metallic tube brushes driven by air, or mechanical cleaners like the Roto or Lagonda can be used during brief shut-downs, or operators may use the Worthington tube washer, which cleans while condenser is in service.

The author assumes that tube loss is reduced by using small condensers. Experience shows that skimping the surface increases the loss which is especially severe in the case of single-pass condensers. Most losses are from corrosion accelerated by temperature to which only the return-flow tubes of two-pass condensers are subjected but to which at least three-quarters of the single-pass tubes are exposed at their outlet ends.

The external air cooler, approximately as shown by the author and such as may be witnessed upon the 17,000-ft. condensers serving the vertical turbines at Port Norris and Yonkers was a Worthington feature until 1905. We found the internal, wedge-shape type, full-length cooler with gradual reduction of cross-section in direction of flow superior to the external type and adopted it.

The single-pass, high-tube-velocity condenser requires excessive circulating water. It is an old trick of the condenser salesman to underbid by offering a small condenser requiring more circulating water. The high-velocity condenser will, the writer admits, show a somewhat higher heat transfer, but when all factors including piping and tunnel cost are properly considered, it does not usually pass the test of dollars by which sound engineering must be proven.

THE AUTHOR. In presenting a subject like this to the Society all the author can be expected to do is to present the test data on the representative size unit in a power plant of the kind Mr. Ricketts is connected with. The actual test data which have been obtained at great expense, are, after all, the property of the Ingersoll-Rand Company, and there are things that are too valuable to put down in a paper of this kind.

As Mr. Reynolds remarks, "The seemingly aimless course of change or improvement in condenser design has been due to the difficulties and expense involved in making condenser tests and in altering existing designs." It was necessary, therefore, for a manufacturer having large facilities to undertake any comprehensive tests and research. But Mr. Reynolds, in common with a number of others, fears that the elimination of waste tubes means a loss of reserve under commercial conditions when the tubes foul. Of all the features of condenser-plant design this one of insuring sustained vacuum is the most important and often the least considered. Condensers and water pumps are proportioned as though clean, soft drinking water were the circulating medium, which is seldom the case.

Excess tubes which are normally waste tubes do not provide reserve against fall of vacuum due to fouling. When tubes foul and the vacuum falls the conditions prevailing on the steam side are very nearly the same as though the injection water were hotter and the vacuum reduced for that reason. It is well known that a condenser which gives 28.5 in. vacuum (91.7 deg.) with 70-deg. injection will give 28 in. vacuum (101.2 deg.) with 80-deg. water, or a fall of vacuum equivalent to the increase in temperature of the water. No claim would be made for waste tubes acting as reserve under these conditions although it is just as important to maintain high vacuum with hot injection as with fouled tubes.

The best reserve against warm water is an ample water supply, and the best reserve against fouling is ample water and high velocities. The trouble arises at the inside of the tube and must be counteracted there.

Mr. Christie points out that a better understanding of corrosion would be of great value. The author believes that there are certain causes of corrosion which can be alleviated by the condenser designer. It is widely known from both practical and laboratory experience that heat accelerates corrosion, and, therefore, the use of higher water velocities and higher water quantities will retard corrosion by keeping the tubes cooler. It is also known that corrosion of the local type takes place where foreign matter deposits in the tubes so that keeping the tubes clean, as by the use of high velocities, will alleviate this form of corrosion. A third form of corrosion that can be controlled by the designer is the water line effect taking place near the inlet end of the tube. At the inlet end there is a contraction due to the more or less square mouthed ferrule

and also a hydraulic disturbance due to the notches in the ferrule. This causes a high velocity contracted inlet with liberation of air and foaming and corrosion similar to the "water-line effect" on ships. This condition is corrected by the use of a fully rounded inlet without notches and with a smooth full-bore entry.

Mr. Christie asks what are the best velocities to use for economical performance. This depends on the kind of water to be passed through the tubes, and it is manifest that if a condenser is to have clear drinking water for the circulating medium the balance of operating versus fixed charges and maintenance cost will come at a lower velocity than where the circulating water causes fouling and corrosion. This question of water velocity has confronted engineers for years, and many large condensers have been installed with velocities of 7 to 8 ft. per sec. whereas others have velocities around 5 ft. per sec.

With a single-pass construction it is possible to get high velocities with low friction head, and it should also be mentioned that the hydraulic circuit is better when the water is lifted to the top of the syphon and goes through the condenser directly, as in a single-pass construction.

Mr. Christie also asks regarding the heat transfer rates obtained. Higher average heat transfer rates are obtained in this condenser because wasted tubes are eliminated. There are undoubtedly many individual tubes in any type of condenser working at high rates, and others working at very poor rates. The purpose of the Ingersoll-Rand design is to eliminate the tubes which are working at very low rates. The author has purposely avoided any statements of heat transfer rates or coefficients because these factors vary widely with the conditions of design.

Mr. Christie's next question is on the desirability of increasing the terminal difference between steam and leaving water in older condensers in order to reduce corrosion. The author believes that in many cases this pays even at the expense of extra circulating pump power, and as pointed out above there are many engineers who have the same belief and use a greater water quantity and higher velocities for a given condenser. In other words a marked difference of opinion and practice already exists as to the desirable water quantity and velocity with any type of condenser. It is the author's opinion that the trend is toward higher and higher velocities.

Mr. Christie also asks regarding the condensate temperature

at partial loads. The full load on a condenser is hard to define because what would be called the full load for a 28.5-in. vacuum rating is only partial load for 28-in. vacuum rating. The question of condensate temperature depends on the proportions of design and on the air leakage and vacuum pump capacity. Hot condensate can be obtained through a wide range of loads, although, as Mr. Christie infers, it is easier to obtain hot condensate at heavy loads and low vacuum than at light loads and high vacuum.

Answering Mr. Ricketts' remarks, the purpose of the present design is to make the majority of the tubes work at nearly full rate instead of having only part of the tubes carrying their full load and others doing nothing. This effective utilization of tubes would not be desirable unless the saving of tube troubles resulted in a condenser more economical of operation and maintenance.

Mr. Ricketts' report on slime formation does not seem to agree with the bacteriological report obtained by the author who understood that slime would not form appreciably in the presence of copper sulphate.

Mr. Grace takes issue with the author on a number of features of condenser design. He is obviously in favor of low velocities, lower in fact than used by many operating companies. The author agrees entirely with Mr. Grace on the merits of low velocity where clear drinking water is to be circulated through the tubes, but disagrees on applying the same basis of design under the widely prevalent conditions of impure water.

Mr. Dean brings up the same point and the danger of too high a cost for circulating the water. The author also refers Mr. Dean to the present existing discrepancy of about 100 per cent in relative circulating-pump power in different plants.

Mr. Dean raises an interesting point regarding condenser performance, namely the action under light loads and with cold water. This is the reason why the author advocates the water distribution shown in Fig. 12, so that in the winter time with reduced quantity of cold water the velocities in the bottom of the condenser can be made low or else a whole section of the condenser shut off entirely.

Mr. Dean and the author are evidently not in accord on the merits of a design using steam lanes. He also takes issue with the author for emphasizing the necessity of extreme care in determining the transverse cross-section and then neglecting the necessity for care in obtaining longitudinal distribution. This feature of design is of great importance but is entirely within the control of the designer.

No. 1820

ECONOMY IN TEXTILE DRYING

By B. R. Andrews, Boston, Mass.
Associate-Member of the Society

The following paper reviews the well-known facts of drying, pointing out—

1 *The desirability of reducing the drying time in any drier at the expense of steam, or even labor, because of the preponderance of the overhead item.*

2 *That this increase in the rate of drying must usually be obtained by increased temperature of the material being dried, and decreased vapor tension of the air actually in contact with it.*

3 *That reduced labor costs, while usually a natural consequence of increased speed from the drier, can often be further reduced by combining the drier with other machinery into a range.*

4 *That steam consumption can be cut only by eliminating leakage, reducing radiation by insulation, and using the minumum amount of new air for removal of water.*

5 *Because cans meet all these conditions, they should be used in preference to all other types of driers wherever the nature of the product will permit.*

THIS paper summarizes some of the means by which those expenses which contribute to the cost of textile drying may be reduced without causing simultaneous increases in damaged goods and without reduction in quality of work produced. There are many such items of expense, but they may be included under the three groups of labor, overhead, and steam.

2 The relative values of these groups are shown approximately in Fig. 1. The labor item includes only that labor actually applied to manufacture; the overhead item includes the labor for maintenance, as well as interest, depreciation, taxes, and the cost of power. While the relative value of each very naturally varies with the different classes of work, most textile drying is done either on cans or in an air drier, so that a fair average is obtained from a range consisting of starch mangles, drycans and tenters, and it was on such a range that the data for plotting Fig. 1 were obtained. Selection was made, not of an average installation, but of one operating with

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a comparatively high efficiency, thereby representing a good distribution of these expenses for low cost of drying.

3 It is natural to attempt drying economies by considering each item separately, yet it is no more possible to do so than in any other manufacturing process, for reduction in labor cost will probably be obtained by increase in investment and consequent overhead expense. If it is obtained by increasing the rate of drying, without much increase in investment, then a sacrifice of steam economy is likely to result. However, if the cost of handling is halved by halving the drying time, with a reasonable increase in overhead

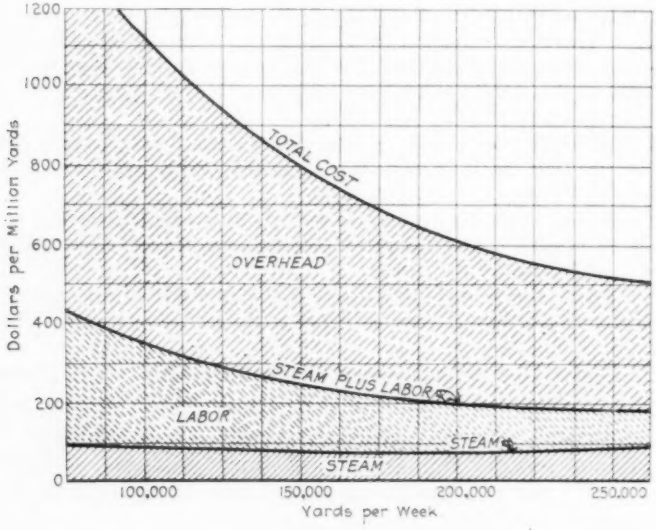


FIG. 1 COST OF DRYING AFTER STARCHING. — 5-YD. GOODS

and no reduction (or even a 50 per cent increase) in steam consumption per pound of goods produced, an overall economy has resulted. Also, the increase of the rate of drying may make possible the combination into a range with calenders, starch mangles, etc., with a reduction of labor, not to one-half, but perhaps to one-third or one-fourth. Therefore, it is usually safe to assume that what is wanted above all else is to secure the maximum speed at which goods can be handled without damage, and then to maintain this speed at the minimum overhead expense and steam consumption.

4 While these conclusions are almost obvious, the actual

drying process itself is not quite so clear, and a general understanding of it is necessary in order to consider this item of speed intelligently. Textile drying is usually nothing more than the conversion from liquid to gas of the free water, or free solution, carried by the material after leaving the extractor, suction box, or squeeze rolls. The hygroscopic moisture enters only rarely into the problem; that is, the process is really "drying," and not the processing often erroneously called "drying," which must be carried on under the most carefully regulated combinations of temperature, humidity and time. Consequently, with drier temperatures not exceeding 220 deg. fahr. and humidities approximating bone-dry atmosphere, we have simply a matching of vapor tension between the water in the material and the air in immediate contact with it. The higher the former, and the lower the latter, the more rapidly the process can be carried on.

5 There are three good ways of raising the vapor tension of the water in the material: (*a*) by contact with a hot surface; (*b*) by contact with hot air having a high wet-bulb temperature; and (*c*) by radiant heat.

6 There are two ways of maintaining a low humidity in the air in immediate contact with the material: (*a*) by a high drier temperature; and (*b*) by rapid air circulation. The second is of the utmost importance because the film of air in immediate contact with the surface in an undisturbed atmosphere becomes very nearly saturated.

7 In the case of a material brought into contact with the hot surface, moisture may be transferred immediately into water vapor by maintaining the temperature of this surface high enough to keep the water in the material above 212 deg., the vapor then condensing and being reëvaporated by the surrounding atmosphere. It is just this method of drying that is utilized with drycans and drum slashers. It gives the most rapid rate of drying, as well as the greatest economy, particularly where an air current is used to disturb the sluggish film of saturated vapor on the surface of the material, thus giving a direct evaporation in addition. Such air circulation is best obtained by properly designed hoods over the slasher and blast nozzles on drycans.

8 In case of a material which cannot be brought into contact with the hot surface, but which must be dried quickly, rapid air currents become almost absolutely necessary and radiant heat desirable. However, the use of air currents and radiant heat is usually

limited by the nature of the material and its possible arrangement in the drier, so that the most common form of textile drying, next to drycans and slashers, is that in which the material is heated and the latent heat of evaporation entirely supplied by the air in contact with it. For example, skein driers and festoon driers, with which the air is blown over or around the product, stock driers, and the "mast" type of knit-goods driers, with which the air is blown through the stock. However, there are cases in which strong air blasts and radiant heat can both be used, such as with a straightaway tenter frame. Here radiant heat alone, plus the convection, which is a natural result, will give one rate of drying; air blasting alone at a high temperature will give a higher rate; but a combination of the two, the highest rate.

9 From the foregoing it will be seen that textile drying resolves itself into two general classes: (*a*) with the material in contact with a hot surface; and (*b*) with the material in contact with air only, so that for a study of steam economy it is best that the two classes be considered separately, and the following table, prepared from the averages of many tests run on efficient drycans without air blast gives the approximate distribution in per cent of the total steam used for the evaporation of one pound of water:

	Per Cent
Loss by radiation	20
Heating wet material	1
Evaporation of water	79
Total	100

or 1225 B.t.u. or $1\frac{1}{4}$ lb. steam at 5 lb. gage.

10 That drying on cans is a very efficient method is apparent and well known. However, actual mill averages often run double this consumption because of time lost between runs and improper proportion of width of goods to the width of cans.

11 In the second class of driers, possible economy of steam is much greater, for while the steam used per pound of water evaporated in a practical air drier can never be equal to that used in contact driers, the average air drier is more complicated and therefore more difficult to build and operate efficiently. The following table gives the destinations of heat required to evaporate one pound of water as calculated from test data on such a drier of more than average efficiency:

	Per Cent
Loss by radiation	14
Heating wet goods	0.5
Evaporation of water	46.5
Air change	39
Total	100

or 2150 B.t.u. or $2\frac{1}{4}$ lb. steam.

12 No loss has been attributed to leakage, because a properly designed drier can usually be so arranged that even though it may not be made air-tight, and openings must be provided for ingress and egress of the material, all of the leakage can be inward and become the new air required for evaporation without upsetting to any great degree the most efficient route of air travel through the drier.

13 Inasmuch as neither the heat required for evaporation, nor that necessary to raise the wet material to the temperature at which evaporation takes place, can be reduced to any material extent, steam economy must be effected by a reduction of radiation loss and the heat required to raise the temperature of the new air used to carry off the water vapor. It is sometimes possible to use the heat contained in this exhausted air either for heating or to augment some other drying process where a high percentage of humidity is permissible or desirable.

DISCUSSION

HALL A. MARION [1] said that the author had pointed out very clearly that the way to increase economy in textile drying was to increase capacity. Several means of increasing capacity had occurred to him. The most important factor was the operator. Many existing installations would be capable of greater capacity if operators could be induced to obtain it. A second means was to improve the load factor in the case of multiple width cans; to load these cans to the proper width with the number of strands for which they were designed. Why should finishing plants not be run twenty-four hours a day like paper mills? Another means of increasing capacity, he said, was to put all machinery in first-class condition, and another by removing the moist air from the cloth.

L. B. MCMILLAN discussed some statements by the author which had to do with loss of heat from galvanized iron surfaces.

[1] 289 Charles St., Providence, R. I.

He pointed out that the tests made at the University of Illinois which showed that more heat was lost from such surfaces when covered with asbestos paper, and to which the author referred, were correct, but he was afraid that a wrong interpretation might be placed upon the results because of the fact that the radiation from bright polished surfaces of galvanized iron or tin would increase as the surface became dulled by rust.

He also spoke of the insulating value of air spaces, pointing out that in order to be effective these spaces must be extremely small so as to prevent a circulation of air within them.

THE AUTHOR replied that he did not wish anything he had said to be taken as a complaint against asbestos insulation on the surfaces he had referred to. It was interesting to note, however, that these surfaces in connection with dryers did stay bright.

J. A. CAMPBELL asked if double ends were still used on dryers to take off the large amount of moisture that comes from nozzles on straight-away tenters. The author replied that he had not seen such installations in the last few years.

CHARLES H. BIGELOW asked what differences were found in applying air on the top and on the bottom of goods in straight-away tenters where a source of hot air was used.

THE AUTHOR replied that such differences would depend upon the arrangement of the heater. It seems to make no difference if only air is used whether it is blown on the top or the bottom but with heater coils located above the goods there is always a danger of spots from dust and water dropping from above.

W. H. CARRIER said he wished to commend the author's consideration of the economic as well as the engineering features of drying.

No. 1821

HIDDEN WASTE IN TEXTILE PLANTS

By Thayer P. Gates, Providence, R. I.
Member of the Society

The purpose of this paper is to point out a few of the losses found in the average textile plant which have been very largely overlooked and which can be greatly reduced through the application of engineering research. The types of losses considered are mainly those occurring in the operation of equipment and the utilization of materials. The paper also points out the opportunities in the textile field for men of engineering training.

IN our textile industries the increase of production per man or per machine during the past twenty-five years has been small as compared with the increase in output from machine tools, and even today most of our manufacturers think of any addition in output only in terms of larger plants. One reason for this condition may be that the operation of our textile plants has been largely in the hands of men who learned most of what they know about the technique of manufacturing within the plants themselves. Many present practices have therefore been based on tradition rather than knowledge of facts, and improvements have been largely along lines of small refinements on present machines. Another result of this type of background is that there has perhaps been less of the professional attitude among textile men than among men in many other lines, and the tendency has been to keep every improvement secret from competitors. A freer interchange of ideas would undoubtedly have resulted in a benefit to all concerned.

2 The graduates of our engineering schools have spent a great deal of time in the study of mill power plants, electric drives, heat balance, etc., all of which play but a small part in the cost of textile products, while a few of them have given much thought to the much more important subject of the improvement of the manufacturing processes and machinery in this industry. During the last few years, however, there has been some change in this condition and we now

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find that a number of textile companies have established research laboratories, or employ trained engineers who are applying critical study and exhaustive research to manufacturing materials, processes and equipment. Such study may result in radical changes in layout, equipment or process, which will vitally affect the textile plants of the future.

3 It is still too soon to predict how rapidly progress will be made, but a great deal may be expected from the engineering approach which respects what has been done in the past but accepts no practice as final until all facts have been collected and examined in the light of a scientific training.

4 The purpose of this paper is to point out a few of the losses found in the average textile plant which have been very largely overlooked and which can be greatly reduced through the application of engineering research. The types of losses considered are mainly those occurring in the operation of equipment and the utilization of materials. The paper also points out the opportunities in the textile field for men of engineering training.

5 The essentials in any study on the elimination of waste are:

a to determine the facts of present operation
b to determine what new facts may be found by analysis or research
c to apply these facts to the improvement of operation.

EXAMPLES OF METHODS EMPLOYED IN ELIMINATION OF WASTE IN TEXTILE MILLS

6 As an example of this method in the study of the operation of equipment, the output from a water mangle as used in cloth-finishing plants may be used. Knowing the present output of the machine, the first step would be to determine the maximum speed at which the machine would operate mechanically, allowing of course a proper factor of safety at the weakest point, and then find whether the goods could be put through the machine at that speed consistent with proper quality.

7 In such a study it may be found that the cloth will not open to width from the rope form at the required speed because the beaters on the scutcher will not take care of it properly. This involves experimental study by the engineer to determine changes necessary on this scutcher to take the increased output.

8 Then it may be found that the goods are damaged on the

expander. The questions are immediately asked: at what place on the expander? what are the reasons for the damage? and how can the difficulty be eliminated? Another job for the engineer. Further, it may be found that the plaiter will not deliver the goods into the truck at the new speed. The engineer again asks why, and by proceeding to overcome this difficulty by employing the methods of engineering research to solve these problems, one at a time, the maximum capacity of the machine is finally reached. In many instances the output as a result of these studies is more than doubled, and the failure to take advantage of this possible increase constitutes a larger "hidden waste."

9 In many plants chemical problems may limit the output. As it is usually possible to redesign the mechanical equipment to take care of the maximum output from the chemical standpoint, the first research work should be done in the chemical laboratory. The problem is to determine what are the variables, and then to solve these variables for the best result of each when combined with each other. This involves the determination of a unit which must be applied in measuring and standardizing each variable.

10 For example, in the determination of luster on a mercerized fabric the standard unit should be saturation, which is measured by shrinkage of the cloth when submerged in caustic. In bleaching, the standard unit is the degree of whiteness combined with the strength and aging qualities of the goods bleached. The chemical research having been completed, the engineer must take these facts and apply them to the study of the equipment to give the maximum output.

11 The first experience of the author with textile research came when he was engineer of a large finishing plant. One of his problems was to locate a new bleach house to take care of approximately 50 per cent increase in production. A careful study failed to reveal any place where the building could be located and any continuity of the processes be maintained. The present bleach house was then examined to see if the increase could be handled there. The machinery was of modern type, was run at the highest practicable speeds, and could not be rearranged to provide for any additional equipment. An examination of the chemical part of the problem showed, however, that a 50 per cent increase could be taken care of with a slight increase in water and chemicals.

12 The problem was finally solved over protests that it had never been done before, by running two strands of cloth side by

side through the same machinery at the same time instead of one as had been the previous practice. Many experiments were made before this was accomplished successfully, but the new method enabled the plant to secure approximately 50 per cent increase in production with practically the same equipment running at the same speed, with the same amount of labor and with a very slight addition of water and chemicals.

13 This solution caused a very considerable saving in the cost of production and also pointed the way to further investigations throughout the plant, with the result that many processes were revolutionized and large savings made.

14 A short time ago a mill treasurer spoke very contemptuously of the possibilities of engineering research in his plant, stating that he knew the business thoroughly, having started in it as a boy. When asked if he was not using individual "dollies" for scouring worsted cloth before dyeing, he replied that he had about twenty-five of them, each one of which required one operative. He was much surprised when informed that all of these dollies could be replaced by one continuous scouring machine, or at most two, requiring but one operative each and using a smaller amount of soap. Is this not a "hidden waste"?

15 The increase of output in the operation of tenter frames in another plant may be taken as a further illustration. After careful engineering investigation, the dry or hot room was perfected, and the frames readjusted to handle the capacity of the dry room. As a result the output was increased over 150 per cent, with a 50 per cent reduction in labor cost. The saving of labor on this job alone within a few weeks more than offset the entire cost of making the study and of changes on the machines.

16 The foregoing examples have all been taken from dyeing and finishing plants. Turning now to an entirely different line, a knitting mill started about eighteen months ago to study its machines to see if the output could be increased. The net result of much research work on these machines together with a study of the yarn used, was that the output of the entire plant was increased more than 25 per cent with the same equipment. The number of imperfections were reduced very materially, and in spite of the higher wage rate there was a substantial reduction in the labor cost per pound of product.

VARIOUS METHODS OF CUTTING DOWN WASTE

17 Occasionally in the study of equipment it is found that cutting down speed may increase output as well as quality. One illustration of this may be found on pickers in a cotton-spinning mill, where a reduction in the speed of the beaters frequently raises the quality of the product, cuts down waste of materials, and betters the production in the later processes.

18 Considerable saving can oftentimes be made in the purchasing of materials. This requires research to determine if a cheaper product may not be used to replace a more expensive one with equal or better results. An example of this is found in the studies made to determine if a lower-grade cotton may not be utilized to advantage. The use of this cotton calls for greater care in purchasing and a careful study of the processes of cleaning and mixing to maintain proper strength and uniformity. This, however, has been done by a number of mills without lowering the quality of their product and has resulted in large savings in the finished product.

19 Another source of loss in many plants is due to the waste of material in process and the reworking of materials which have become damaged. Records should be kept to show the extent of these losses as such records will undoubtedly assist in reducing the waste and act as a stimulant to maintain them at a minimum. In some finishing plants, for example, the amount of rehandled goods due to stains, poor dyeing, poor bleach, etc., oftentimes reaches 15 per cent of the work done, while in other plants it falls below 1 per cent. Frequently the amounts of seconds, remnants and rags reach large figures. In some plants no records are maintained of such losses, so undoubtedly these can be characterized as "hidden wastes."

20 In addition to the studies in the operation of the machines and in the utilization of materials, it is desirable to standardize thoroughly the labor jobs on these machines. Frequently the engineer can develop labor-saving devices that will reduce the number of operatives required. In other cases standardization may be carried to such an extent that less skill is needed on the part of the operatives and lower-grade labor can be used.

21 Throughout this paper emphasis has been placed on the savings which may be made by engineering research in the textile field, which has been much neglected up to this time, and it is the

belief of the author that in the future great improvements in operation will undoubtedly result by the replacing of tradition with definite knowledge of facts determined by such research.

DISCUSSION

CLARENCE W. MARSH spoke of the possibilities of textile plants making some of the materials used by them. It is a fact, he said, that any concern using chlorine and caustic soda, especially a textile mill where steam is used for heating and power is therefore comparatively cheap, can make it infinitely cheaper than it can be bought. The reason, he said, is that many of the steps necessary to convert chlorine bleach into the finished product which the textile manufacturer uses represent economic waste. Chlorine and caustic soda can be used as delivered from the mill without conversion. To liquefy chlorine, he said, costs a cent and a half, the container from half a cent to a cent, and shipping a cent and a half. Money must be spent on the control apparatus for the liquid chlorine when an equal amount might purchase apparatus for making chlorine. And finally, the liquid chlorine must be transformed again into a gas.

FRANCIS A. CHIFFELLE said he wanted to emphasize the need of plant investigation. He asked if there was not a limit to the speed of tenters. A tenter chain, he said, was not built for high speed. Chain manufacturers would hesitate about putting in a chain like a tenter chain on ten-tooth sprockets and yet all tenter chain is run on ten-tooth sprockets. With expensive chain like tenter chain there is a limit to the speed which should be allowed. Chain manufacturers, he said, figure on about 350 ft. or 116 yards per minute. The author said he had known tenter chains which had been running two years at a speed of 110 yards per minute, and occasionally at a maximum of 130 yards per minute. Mr. Chiffelle thought that this speed was too great for economy, and the author replied that the problem was to determine whether it was cheaper to run fast and have the chain wear out in a few years, or to run slow and have it last twenty.

D. M. BATES spoke of his experience in replacing the iron bearings of chains with maple, and of work done by the late H. L. Gantt in doubling the yardage in textile plants with less wear and

tear on the machinery. In regard to textile mills making their own bleach, as had been suggested by Mr. Marsh, he pointed out that this had been considered many times. The convenience of handling the liquid chlorine was appealing, even though the liquid was expensive. There were also difficulties, he said, in making chlorine in the relatively small quantities required by the average cotton bleachery.

H. M. BURKE said that as an engineer he realized that more scientific knowledge of the textile industry was needed. It was hard, he said, to convince owners of textile plants that this was so. Much of the knowledge of methods and machinery in finishing plants, however, could be standardized. There was, he thought, a need for research in this field. It might be necessary to get enough publicity from engineers to sell such ideas to the man handling the money in a textile plant. This could not be done properly unless the processes were studied scientifically.

He compared the textile with the shoe industry in which highly specialized machinery had been introduced as a result of study.

THE AUTHOR said that it had been his experience that it was easier to convince the man who handles the money than a board of directors. He had found it a good plan to tackle one problem at a time and to get results. Results were always pleasing to the board of directors.

No. 1822

PREVENTION OF WASTES IN INDUSTRY

By Fred J. Miller, New York, N. Y.
Member of the Society

I AM asked to discuss, not simply wastes in industry, but incentives for the prevention of such wastes, and here we are confronted with the anomaly that those who would be supposed to have the greater incentive for the prevention of industrial wastes are found to be responsible for the major portion of the wastes that occur. Responsible, I mean, in the sense that they, the owners and directors of industries, are the ones and the only ones, who can adopt effective means to stop these wastes.

This seems to indicate that not only must there be incentives for the avoidance of waste, but there must be also a clear realization that preventable wastes are occurring and a general understanding of the reasons for them and of the means by which they may be prevented.

It is natural for the normal man to be active in some way — either mentally or physically, or both. There is scarcely a greater punishment that can be inflicted upon him than complete, enforced idleness, such as results from solitary confinement in a dark cell.

It is perhaps industry's chief problem to conserve, develop and make use of the natural desire of the normal individual to be accomplishing something, and, further, to relieve workers so far as possible from such conditions of work as are deadening to ambition, to initiative and to the creative instinct. Far more can be done along that line than may seem possible at first sight, and the results of even the simplest efforts in that direction have, in many cases, been very excellent for all concerned.

INCENTIVES IN INDUSTRY DEFINED

Incentives may be grouped in two general classes which may be called the "penalty" class and the "reward" class.

Part of an address delivered at the session on Elimination of Waste in Industry of the Annual Meeting, December 1921, of The American Society of Mechanical Engineers.

The penalty incentives, in the form of the lash or other gross physical punishments, were more common in slave times than they are now, but the penalty incentive still persists here and there in various forms and there are still too many in our industries who seem to recognize no other kind; not knowing it is still true that "he who owns a slave is himself in chains."

Incentives of the penalty class make people afraid not to do the things they are ordered to do. Incentives of the reward class tend to make people want to do the things that are expected of them. These two classes of incentives of course differ widely in their nature; but they differ no more than the results that are obtained by them.

I do not know what course the discussion of this question may take here today, but I prefer to confine myself to discussion of the reward class of incentives as being the only one worthy of consideration in connection with industries carried on in an enlightened age and in dealing with free men.

The reward class of incentives may be divided into two subclasses that may be called respectively "group" rewards and "individual" rewards.

Profit sharing, group insurance, employee representation or participation in management, most of the so-called welfare work, general or so-called horizontal increases in wages, bonuses paid to all employees alike, etc., are and are intended to be group incentives; while individual increase of wages, piece work, bonuses to individuals for specific individual attainments are, of course, individual incentives.

Whether group or individual incentives should be employed in a given case depends of course upon the circumstances of that case; but my own experience and observation lead me to the belief that, after we have established good working conditions, such as well-lighted, heated and ventilated work places; done what we can to keep them in good sanitary condition and as free as possible from danger of accidents; and have foremen and other executives who have been selected and trained to take an enlightened and "human" attitude toward employees, the rest of the way to the best possible general results is through individual reward incentives; which may take the form of higher hourly wages paid to individuals for individual attainments, a bonus paid for definite attainment, advancement from the ranks to successively higher executive positions, or all of these together with, so far as I have

been able to perceive, about equally good results when applied intelligently. And they can be applied intelligently only when means are provided for having a continuous record of every employee's performance with reference to established standards; so that reward incentives will not be based upon any executive's prejudices or whims but upon actual and demonstrated service rendered in doing the work for which the plant is operated.

As a consequence of the acute situation and the very troublous times we have been, and still are, passing through, there has been, I think, some tendency to overelaboration in certain of the measures taken to overcome our industrial difficulties. As for me, I still have faith in the comparatively simple means of enlisting the enthusiastic coöperation of minor executives and employees, and an essential part of this is to give them such treatment as every man likes to receive from those with whom he comes in contact; always remembering that workpeople are not, after all, essentially different from other people; are at least as readily responsive to candid, fair and courteous treatment as are other people, and also as well able to judge whether or not they are receiving it.

CAUSES OF INDUSTRIAL WASTES ITEMIZED

And along with this there must be avoidance of the large industrial wastes that come from overloaded inventories; slow movement of materials through the successive operations of manufacturing; unskilled, because inadequately studied, and developed, manipulation of materials; inability to definitely and promptly place responsibility for delays; failure to clearly distinguish between those things which are the worker's responsibility, and the foreman's responsibility, the superintendent's responsibility, and the owner's responsibility.

That all these, as well as the designing engineer, have their separate responsibilities is, in a general way, well recognized; but in our manufacturing establishments there is usually no means for definitely assigning responsibility in such manner that a record is made, clearly showing to all concerned where the responsibility lies and whose duty it is to take steps to correct the defect, assuming it to be remediable. Such a record does away with arguments and the attempts to shift blame from one to another, until finally it rests upon the man who cannot "talk back," or is canny enough not to do so and must bear it, smarting under the belief

that he is unjustly blamed; and being disheartened and perhaps alienated by what he regards as being grossly unfair to him.

The major cause of waste in manufacturing lies in defective administrative methods, for which in general no one is to be seriously blamed, for they are the methods that have the sanction of long usage and by them, or rather in spite of them, many successful enterprises have been, and are being, conducted.

Certainly owners and managers are not to be blamed in the moral sense for following well-established methods and practices, and of course they should follow them until they become convinced that there are better methods and practices that are open to them. On the one hand we see employers who are too easily prevailed upon by charlatans to take up methods that are little better than a group of unrelated "stunts," and on the other hand those who always delay progress along new lines until pretty nearly every one else is far in advance of them.

It is the work of engineers to educate and to show the better ways, and I predict that as time goes on our industries will be more and more directed by engineers who know how to direct them for production and who will regard production and service as the prime objects to be attained by an industrial organization. When this change has been effected, one of the greatest causes of waste will have been removed.

I could mention entire industries that are in the control of men who are little if at all interested in production, but devote their entire attention to high finance, which is too often crooked finance.

HUMAN ELEMENT MUST BE RECOGNIZED

All signs point to new conditions under which our industries must be conducted. Improved methods of administration may go very far, but, however, along with them, and an essential part of them, if we are to attain the highest immediate success and prepare for still further progress in the future, must be full recognition of the transcendent importance of the human element in our industries, and means simply must be found to remedy and avoid the condition into which so many of our industries have fallen and in which the attitude of employers and employees toward each other ranges from indifference or suspicion to more or less open and avowed hostility and a keen desire for revenge of wrongs, real or imaginary.

Ambitious men must be expected to have ambition for the wel-

fare of their families, and from the social, political and industrial standpoints this is desirable and indeed necessary if our industries are to thrive by rendering service to society.

Foremen must be looked upon and must regard themselves as leaders, inspirers and teachers of their men rather than mere drivers; means must be provided by which the value of workers with repect to a fair standard can be indubitably ascertained and each should be unfailingly rewarded in proportion to his attainments.

It must be recognized that even in the simplest work pride of achievement can usually be developed, and that often men will work their best only when they are not hampered by conditions that limit their achievements or that prevent them from being credited with their achievements; especially when they know, or believe, that these hampering conditions can be removed.

It would, I think, surprise a great many of us if we could know just how much of the wastes of industry that are caused by careless, inefficient work have a deeper underlying cause in a feeling created by the conditions under which men work, that make them believe they cannot do what they should do, and for reasons entirely beyond their control.

In a certain government establishment to which many good workers came during the war, mainly for patriotic reasons, quite a number of these men quit because conditions there were such that they could not do a fair day's work, no matter how much they might try. They did as much as was customary there, but so much less than they had been accustomed to do that they became disgusted and could not bear to remain.

It is by no means always easy to say where the blame lies for industrial wastes that come from slackness of workers and executives.

Take the case of a machine shop in which it is apparent that not much more than half the work is turned out that should be done. In many such cases the work has been brought up to the full standard by a change in the management, with or without a change in staff personnel, but employing the same workers as before.

BETTER MANAGEMENT OFTEN REDUCES WASTE

Too many there are who would, with all the assurance in the world, blame the previous inefficiency entirely upon the employees, but the hard fact that such inefficient establishments have been in many cases vastly improved by change of management or of manage-

ment methods must be squarely faced if we are to comprehend our problems or succeed in materially improving conditions.

Especially do we need to adopt such methods of management as will enable the facts to be fairly presented to both sides in every difference that arises between employer and employee; and experience shows that when we have done that, both the employer and the employee are far more reasonable and considerate than either usually imagines the other to be.

It is at least as important that all the elements of an industrial organization should work together harmoniously and without friction as that the different parts of a finely designed and constructed machine should do so. When either does not function properly, it is a case for the use of intelligent discrimination in finding out the real cause of the trouble and the proper remedy. Usually the bludgeon treatment only makes things worse, no matter which side resorts to it.

Production has been and is restricted by workers, both organized and unorganized, and most of such restriction is of course wrong, from the economic standpoint, if not ethically.

In most industries, however, I think it can easily be shown that restriction of production by workers is insignificant compared with the restrictions caused by financial juggling of one kind or another; by avoidable irregularity of employment of labor and of plant by presidents or managers who are temperamentally unable to make decisions and then stick to them, or are unable to do so because they are under the control of men "higher up" who know nothing of industrial science, or even that there is any such thing; by unnecessarily large inventories and consequent tying up of capital that could be otherwise usefully employed; by inadequate control of the movement of material through the works from one operation to another and from one department to another; inadequate, or entire absence of provision for teaching or training operators, and minor executives; absence of effective means of recording attainments of workers, foremen, etc., so that their standing does not so much depend on actual performance as upon other things, some of them little if at all related to the work, such as nationality, religion, membership in secret organizations or fraternities, and sometimes plain graft.

When we set out to discuss an acknowledged fault our picture is necessarily rather a dark one. There is, of course, a bright side. We must work to make that bright side still brighter.

The greatest and most effective incentive for the prevention of industrial wastes is disarmament, so to speak; the cultivation of friendly relations between all those concerned in industrial enterprises; and the maintenance of such a system as will enable every man, from the highest to the lowest, to know what he is responsible for, to whom he is responsible, and that he personally will be credited and rewarded in proportion to service rendered.

No. 1823

THE STRENGTH OF AIRPLANE RIB FORMS

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AND

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Junior-Members of the Society

As illustrating the problem of the strength of materials entering into airplane construction and the degree of research necessary for their proper solution, the following report of an investigation carried out at the United States Forest Products Laboratory will be of interest. In the design of airplane ribs the prime requisite is to obtain the maximum strength with minimum weight of rib. Two factors enter into the problem: the form or design of the rib and the method of its construction in relation to the grain of the material. It will be seen that the investigation reported was conducted in a very complete manner and that conclusions of great value in the design of this essential part of the airplane are brought out.

THE great difference in the strength properties of wood along and across the grain has long been recongized as one of the features restricting its use in structures. Tests made by the U.S. Forest Products Laboratory show that the approximate ratios of the strength properties in these two directions are: The modulus of elasticity across the grain is one-fifteenth of that along the grain; the tensile strength across the grain is one-twentieth of that along the grain. These ratios do not apply exactly to any individual species, since different species vary largely in this respect.

2 A very material reduction in the differences along and across the grain is obtained in ply-wood construction by gluing together successive sheets of veneer with the directions of the grain at right angles, and varying the proportion of grain and number of plies in the two directions.

3 In the design of airplane ribs the prime requisite is to obtain the maximum strength with minimum weight of rib. This series of tests has been made to find the strongest form of rib, taking into ac-

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count the two factors above mentioned by placing the grain of the veneer 45 deg. to the vertical. In modern practice the face grains are vertical and the core horizontal. Most of these tests, however, were made with the face grains parallel and at 45 deg. to the vertical with one core at 90 deg. to the face grain.

DESCRIPTION OF THE MATERIALS USED

4 Birch veneer $\frac{1}{28}$ in. thick was used. In order to obtain a fair comparison in the tests, all of the veneer used in making up the ply-wood was cut from one log. The veneer used consisted entirely of heart wood, the log being sufficiently large to eliminate the need of using any sap wood. No pieces containing knots or checks were used. The veneer had been in store for some time and was well seasoned.

5 The veneer was glued together into three-ply ply-wood by the Theodore Schwamb Company, which specializes in such work. The ply-wood was cut into specimens $4\frac{1}{2}$ in. wide by 18 in. long. As already noted, the face grains were parallel and in most of the tests at 45 deg. to the vertical, and the core was placed at 90 deg. to the face grains. One of the chief objects of these experiments was to determine the effect of making the web of a rib from ply-wood with the grain at 45 deg. to the chord, so that the grain runs parallel to the directions of the principal stresses at the neutral axis of the web.

6 The ply-wood now used in practice in airplane ribs is made up with the face grains parallel and usually vertical and the core horizontal. This method has proved generally stronger than if the reverse disposition is used, that is with the face grains parallel and horizontal and the core vertical.

7 In order to prove the superiority of the method suggested in this paper some specimens were made according to modern practice and tested under like conditions.

THE METHODS OF TESTING EMPLOYED

8 In all cases the specimens were tested as cantilevers at a 15-in. arm. Spruce flanges $\frac{1}{4}$-in. square were glued on the sides of the specimens to prevent lateral collapse as a whole. In order to insure a definite length of cantilever arm and also for holding purposes, a hardwood block, $\frac{1}{4}$-in. by $3\frac{1}{2}$-in. square was glued on each

side at one end of the specimen. The specimens were held in a wide-jawed vise; knurled jaws were used to avoid slippage. The load was applied by a smooth-acting screw jack, and the force was measured by placing the screw jack on a small platform scale which could be read accurately to one pound. Upon the head of the screw

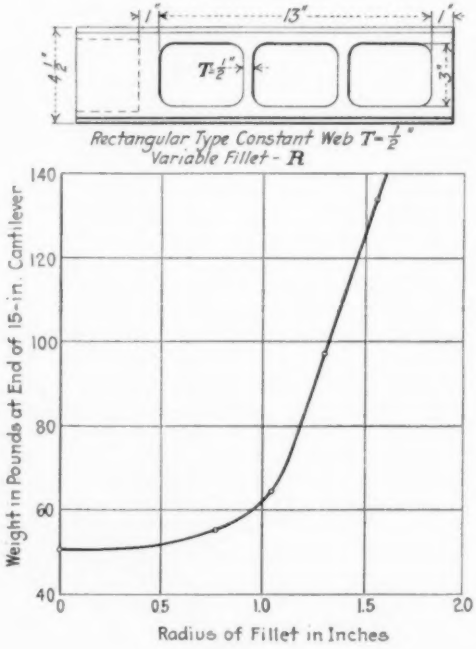


FIG. 1.

jack, a round steel bar was placed. This insured a concentrated load at the end of the cantilever.

THE RESULTS OBTAINED

9 In the rectangular type of lightening hole with constant web thickness and variable fillet radius, it was found that below ¾ in. radius the fillet was of but little use. This can be seen plainly by the curve of Fig. 1. The maximum force applied at the end of a 15-inch cantilever in this group was 134 lb., the fillet radius being 1½ in. thus making the ends of the lightening holes semi-circular. The failure of all the specimens in this group was not a pure shear

but due to beam action in the vertical portions of the web, producing failure by tension cracks.

10 A test specimen was made similar to the one of maximum strength in this group with the face grains parallel and vertical and the core horizontal. As might be expected, this specimen was stronger since the grain was in the proper direction to resist the forces which occur in this form of design. This specimen broke at 181 pounds, showing the vertical grain to be better than the diagonal for this

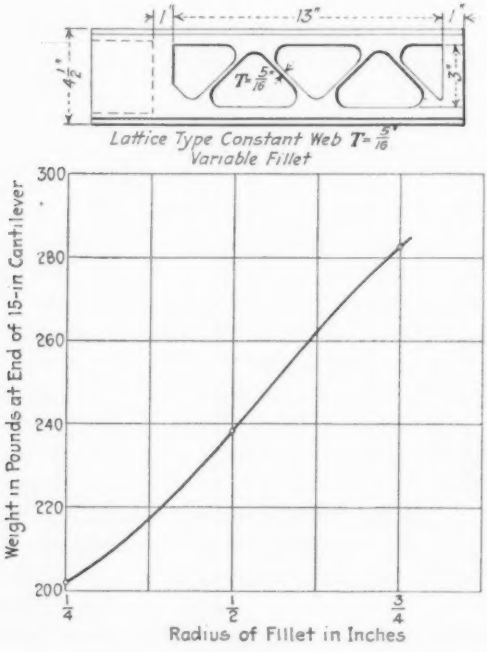


FIG. 2.

particular form of lightening hole. This does not demonstrate any general superiority of the vertical grain, however, since the rectangular form of lightening hole gives a web obviously very ill-suited for showing whatever merits the use of inclined grain in the web may possess.

11 The specimens for the tests made in the next group, as shown in Fig. 2, were made with variable radius of fillet using lattice construction. A maximum of 283 lb. was reached in this group. Since 181 lb. was the maximum force applied at the end of a 15-in.

cantilever with vertical grain and rectangular holes, and since it has generally been found that lattice design with triangular holes is little, if any, stronger than that involving amply filleted rectangular holes, if vertical or horizontal face grain is used in both cases[1] the advantage of diagonal grain design for the lattice form appears to be clear. The comparison between the best case with diagonal grain and lattice design and the best arrangement of rectangular lightening holes with vertical grain expressed in percentage change from the second to the first is as follows:

Radius of fillet, in.	Increase or decrease of area not cut out (change of web weight, per cent)	Increase or decrease of force applied to produce failure, per cent
$\frac{1}{8}$	− 29	+ 12
$\frac{1}{2}$	− 2	+ 29
$\frac{1}{4}$	+ 2	+ 56

The advantage of the lattice construction is obvious. Inspection of Table 1, where the ratio of strength to area of web is given, indicates

TABLE 1 RATIO OF STRENGTH TO AREA OF WEB

Type of Construction	Variable	Width of Vertical or Diagonal Web Members, in	Radius of Fillet, in.	Character of Failure	Distance between Center of Cross	Per Cent of Total Area Cut Out	Per cent of Area Remaining	Force Applied at End of 15-in. Cantilever, lb.	Force ÷ Per cent of Area Remaining
Rectangular lightening holes	Radius of fillet	$\frac{1}{2}$	0	Shear	....	76.1	23.9	50	2.09
		$\frac{1}{2}$	$\frac{1}{4}$	Shear	....	73.1	26.9	55	2.04
		$\frac{1}{2}$	1	Shear	....	70.7	29.3	64	2.18
		$\frac{1}{2}$	$1\frac{1}{4}$	Shear	....	67.7	32.3	97	3.00
		$\frac{1}{2}$	$1\frac{1}{2}$	Shear	....	63.9	36.1	134	3.71
Lattice	Radius of fillet	$\frac{5}{16}$	$\frac{1}{4}$	Compression	....	74.6	25.4	202	7.95
		$\frac{5}{16}$	$\frac{1}{2}$	Compression	....	70.3	29.7	233	7.85
		$\frac{5}{16}$	$\frac{3}{4}$	Compression	....	63.3	36.7	283	7.71
Lattice	Thickness of web	$\frac{5}{16}$	$\frac{1}{4}$	Compression	....	74.6	25.4	202	7.95
		$\frac{3}{8}$	$\frac{1}{4}$	Compression	....	73.0	27.0	238	8.82
		$\frac{7}{16}$	$\frac{1}{4}$	Compression	....	71.5	28.5	256	8.98
		$\frac{1}{2}$	$\frac{1}{4}$	Tension	....	69.9	30.1	278	9.24
Crossed lattice	Distance between center of cross	$\frac{1}{4}$	$\frac{1}{4}$	Compression	7 in.	70.9	29.1	201	6.90
		$\frac{1}{4}$	$\frac{1}{4}$	Compression	6 in.	69.3	30.7	230	7.49
		$\frac{1}{4}$	$\frac{1}{4}$	Compression	5 in.	67.2	32.8	280	8.54
		$\frac{1}{4}$	$\frac{1}{4}$	Flange	4.5 in.	65.7	34.3	295	8.60

[1] Bulletin Airplane Engineering Division, U.S. Army, September, 1918.

that that ratio is little affected by radius of fillet, the smaller radii having a slight advantage.

12 In the next group, shown in Fig. 3, the lattic construction was used with a constant fillet radius and a variable width of web

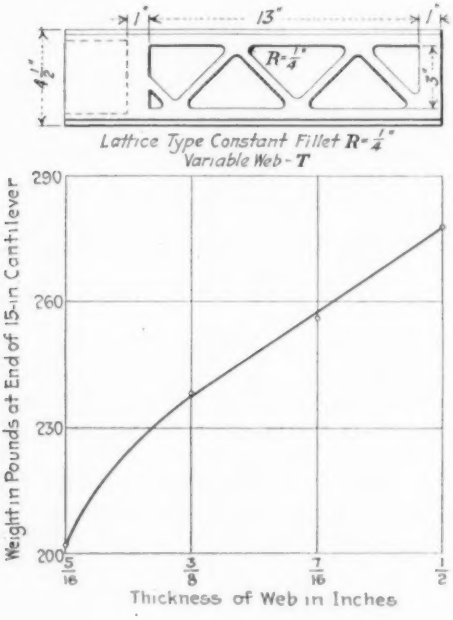


FIG. 3.

diagonal. These tests compare with the rectangular-hole standard as follows:

Width of fillet, in.	Increase or decrease of area not cut out, per cent	Increase or decrease in force applied, per cent
5/16	− 29	+ 12
3/8	− 25	+ 32
7/16	− 21	+ 41
1/2	− 18	+ 54

13 The ratio of strength to weight is steadily improved by increasing the width of the diagonals, although the improvement is low for widths of 3/8 in. or more.

14 The crossed lattice type, as shown in Fig. 4, compares with the standard construction as follows:

Distance between centers, in.	Increase or decrease of area not cut out, per cent	Increase or decrease in force applied, per cent
7	− 19	+ 11
6	− 15	+ 27
5	− 9	+ 55
4.5	− 5	+ 63

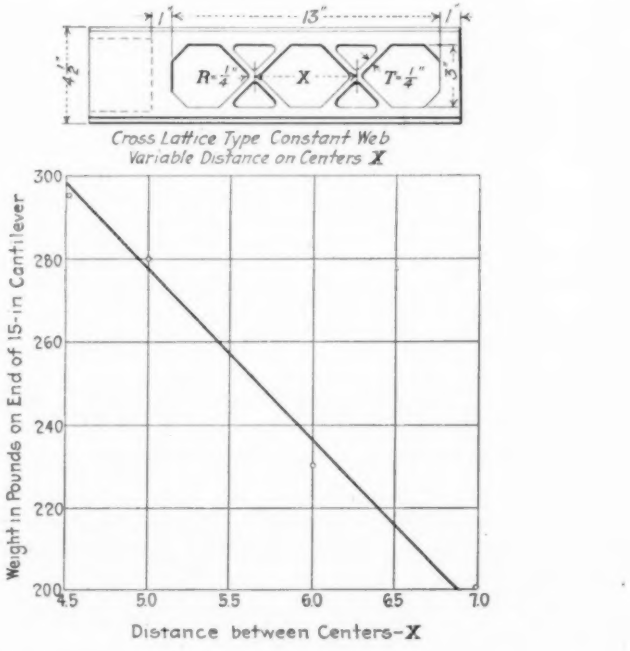


FIG. 4.

15 A crossed-lattice arrangement with a spacing between the crosses of 1.5 times the depth of the lightening hole appears to be approximately equal to the best straight-lattice arrangement. It is difficult to make a satisfactory direct comparison, as the required depth of the horizontal portions of the web, above and below the lightening holes, varies with the distance between lattice members. With due

allowance for this fact, a very close spacing of the crosses seems to be best.

16 With forces in excess of 300 lb. the flange fails by shearing. In the crossed-lattice construction true shear failure of the diagonals never occurred. The compression members are supported by the tension members in such a way as to decrease the free length of the column under compression.

17 In connection with this crossed-lattice group some specimens were tested with the compression member left out in every

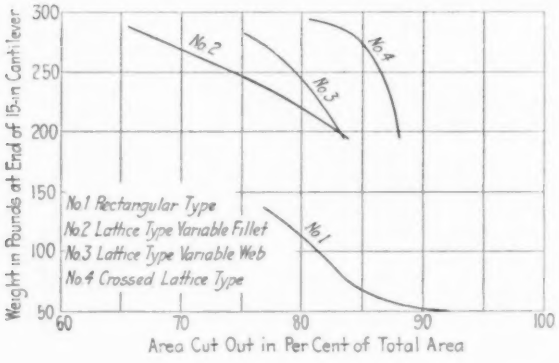


FIG. 5.

other cross. This arrangement is palpably inferior to the straight-lattice or continuous crossed-lattice. It was found that the crossed-lattice type did not stand up so well relatively with web thickness of $\frac{3}{16}$ in.

18 A continuous cross lattice spaced at 3 in. with vertical grain, carried 214 lb. Even this design is far better than that with rectangular holes.

19 Finally a test was made with no area cut out and it failed by buckling of the flange at 325 lb.

20 It was observed in the tests made that there was but little deflection with both the straight-lattice and cross-lattice type. The deflection with the rectangular type was much more pronounced.

21 A comparison of the four types of rib forms is made graphically in Fig. 5.

No. 1824

THE PRESENT TREND OF TURBINE DEVELOPMENT

By Lewis F. Moody, Philadelphia, Pa.
Member of the Society

IN order to gain some idea of the direction in which the development of the hydraulic turbine is tending, it is useful to consider the path by which the turbine has reached its present condition, and a few phases of the turbine's evolution will be briefly mentioned.

2 The early turbines during the first half of the last century were most frequently of the single-runner vertical-shaft type, usually set in open flumes and having simple radial or axial flow types of guide-vanes and runners. Upon the advent of electrical power, the turbine began to receive elaboration, the vertical arrangement of shaft being abandoned and as many as eight runners sometimes being placed in a single turbine. Straight draft tubes were abandoned in favor of the elbow type, which was inherently unsuited to handle a whirling discharge from the runner such as exists in turbines of high speed at all gate openings and at part-gate in all turbines.

3 With the growth of unit capacity, the importance of the turbine and its engineering problems increased correspondingly. When the turbine had reached a size such that any interruption of a single unit's service would involve a loss of ten or twenty thousand horsepower, it no longer became desirable to install a low-grade of machinery nor to place many of the working parts within the water passages where they would be inaccessible for inspection, lubrication, adjustment or repair. The direction of evolution then took the form

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Presented in Philadelphia Jan. 21, 1921, at a Symposium on Hydroelectric Development and Distribution, held under the auspices of the Engineers' Club of Philadelphia and the Philadelphia Sections of the American Society of Civil Engineers, the American Institute of Electrical Engineers and The American Society of Mechanical Engineers. Somewhat condensed. The paper in its complete form can be found in the March 1921 issue of the Journal of the Engineers Club of Philadelphia.

of a return to a simpler machine and it no longer became the style to place four or six runners in a flume, each runner having twenty or more separate movable guide vanes with a large number of the parts of the mechanism for operating them entirely submerged.

4 The general arrangement of the turbine has therefore gone the round of increasing complexity followed by a return to simplicity, and this applies not only to the general arrangement and mechanical features, but to the hydraulic design as well. In place of the simple types of guides and runners, the effort toward increased speeds and capacities resulted in the adoption of so-called "mixed-flow" runners with complicated forms of wheel vanes, the water changing in direction from inward to axial or even to outward flow within the runner. With runners of high specific speed, the runner vanes are cut back to a much smaller radius than that of the ends of the guide vanes in order to avoid excessive vane velocities, and a large transition space exists between the guides and runner. In turbines of extremely high specific speed, there has been a return to simple runner types, including the purely diagonal and axial flow runner. Such types seem the logical development of the process of cutting back more and more the vanes of the high speed "mixed-flow" runner. The trend of development has thus been toward a turbine in which there is a more gradual change in the direction of flow within the runner. This change has simplified both the lines of flow and the forms of the runner vanes.

5 With the adoption of the "mixed-flow" type of turbine, academic methods of calculating the hydraulic elements of the design became less and less applicable and for a number of years hydraulic engineers have been greatly handicapped by the lack of any satisfactory method of rational design due to the complicated form which the turbine has assumed, together with the natural difficulties inherent in practically every hydraulic turbine problem.

6 The writer believes that it will be worth while to give attention to some of the theoretical problems involved in the hydraulics of the modern turbine. Without attempting to develop a complete theory, a few considerations or "speculations" will be explained below, which have a bearing on the further development of the turbine. In particular, it will be of interest to look into the possibilities of securing further increases in specific speed without undue sacrifice of efficiency.

7 The chief elements of the turbine of the present, and probably of the immediate future, are:

a A casing in which the water can approach the runner symmetrically with respect to the axis on all sides, without any obstructions or enforced abrupt changes of flow which would introduce eddies or unsymmetrical distribution of velocity. The spiral or volute casing meets these conditions, and in most cases is the most satisfactory form of intake. On account of the increase in the velocity of the water from a low value in the intake to a high value where it passes through the runner, the water passage must contract radially as it approaches the runner, and must subsequently expand again. The approach of the water to the runner therefore involves a radial component of flow toward the axis; and by so-called "speed vanes" or stay vanes and by the movable guide vanes or gates the water is given increasing whirl or rotational components of motion about the axis as it nears the runner. After leaving the guide vanes the water continues to increase in velocity of whirl due to its closer approach to the axis, and at the same time it turns in the meridian plane and takes the axial direction either just before, during, or just after its passage through the runner.

b A wide transition space is provided between guide vanes and wheel vanes, and in this space the water can come together in a continuous whirling mass before entering the runner. In a high-specific-speed turbine the water continues to contain whirl components of motion, but of somewhat reduced amount, after leaving the runner, even when the turbine is operating at its point of best efficiency.

c After leaving the runner the water continues to flow as a whirling, axially advancing stream in a draft tube having a straight axis coincident with that of the turbine, and conical or flaring walls, this form of tube continuing, if what the writer believes to be the best practice is followed, until the velocity head has been reduced to so low a value that no serious disturbance in the symmetry of flow through the turbine, or material loss of head, will be caused by diverting the flow into other directions or by finally discharging the water.

8 . The turbine may be viewed as a water passage having walls which are surfaces of revolution and containing a continuous body of water which is both whirling and progressing with radial and axial velocity components, in which is placed a runner which develops torque by reducing, but not necessarily destroying, the whirl components of the flow. If a draft tube is used which is capable of regaining a considerable part of the energy due to the whirl compo-

nents as well as that due to the axial components of the velocity of discharge from the runner, a result which can actually be secured, the "outflow loss" from the runner is reduced to a small fraction of that which enters into the usual textbook formulas.

9 The conditions of flow in the transition space in advance of the runner and in the draft tube, as well as the action of the water

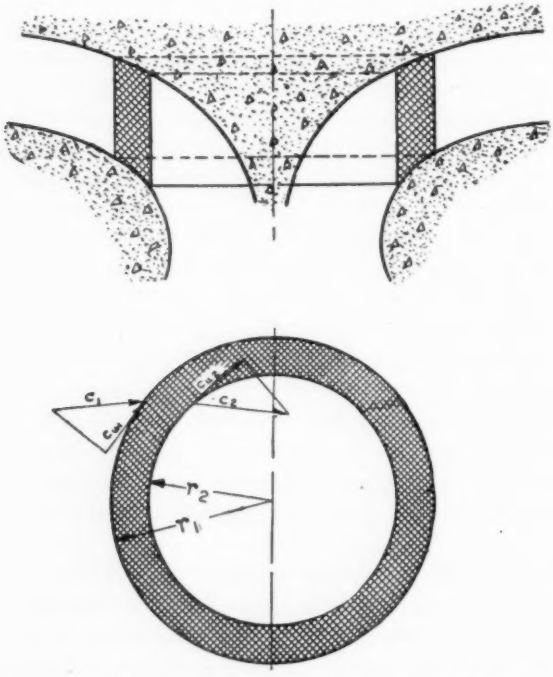


FIG. 1 FLOW IN TRANSITION SPACE IN ADVANCE OF RUNNER AND IN DRAFT TUBE

on the runner, can be usefully treated by means of the following method:

10 Consider the steady flow of water through a passage (Fig. 1) bounded by surfaces of revolution, and imagine the water to be given an initial whirl either by guide vanes or volute casing or equivalent means. Let us center our attention upon the annular space marked by double hatching. This ring is continually filled with water which is flowing in through its outer cylindrical surface of radius r_1 and out through its inner surface of radius r_2; and at any

particular point in the space the velocity, pressure, etc., will be the same from instant to instant, since steady flow is being considered. There will therefore be no change with respect to time in the conditions within the ring-shaped space; so that this ring of water is in equilibrium, and is acted on by balanced forces, with zero acceleration.

11 If the water enters the space with velocity c_1, having a circumferential component c_{u1}, and leaves with velocity c_2 having a circumferential component c_{u2}, then, applying the well-known principle that the "impulse" or force exerted by W lb. of water of velocity V is WV/g, we shall have the moment exerted upon the ring of water by the entering streams, about the axis, equal to $Wc_{u1}r_1/g$, in which W is the total weight of water entering the ring per second. In the same manner the moment due to the backward reaction of the leaving stream is $Wc_{u2}r_2/g$, the amount of water leaving being the same as that entering in accordance with continuity of flow. The two moments must be equal and opposite, since otherwise the ring of water would indefinitely increase in velocity; so that neglecting the frictional resistance of the walls, we have for steady flow

$$\frac{Wc_{u1}r_1}{g} = \frac{Wc_{u2}r_2}{g}$$

or

$$r_1c_{u1} = r_2c_{u2}$$

This relation will apply in any unobstructed space of revolution, whether the flow is inward, axial, or outward. The principle is not new. It was known in the case of a vortex to Leonardo da Vinci.[1]

12 The above relation means that in the free flow within such a space of revolution the velocity of whirl varies inversely as the radius. We can illustrate the use of this law by applying it to the familiar case of a free vortex in an open basin. If we neglect the components of velocity in the meridian plane and assume for an approximation that the velocity is entirely in the circumferential direction we shall have, referring to Fig. 2, $v = k/r$, r being the radius to any point in the surface and k a constant; and if z is the ordinate of the same point, then since the pressure at every point of the surface is atmospheric, and since therefore the velocity head must increase by the same amount that the elevation head decreases, we shall have $z = v^2/2g$. Combining the above expressions we have

[1] Forchheimer: Hydraulik, p. 285.

r^2z = (a constant). This is the equation of a third-degree hyperbola, which agrees closely with the surface of a free vortex.

13 It is interesting to note that where there is a whirl component of velocity in the flow in such spaces of revolution there must be a surface of discontinuity bounding the flowing stream, similar to the surface of the above vortex, since if the flow should approach the axis more closely the velocity of whirl would be required by the above relation to approach infinite values. When the space within the surface of discontinuity is filled with water it is likely that this

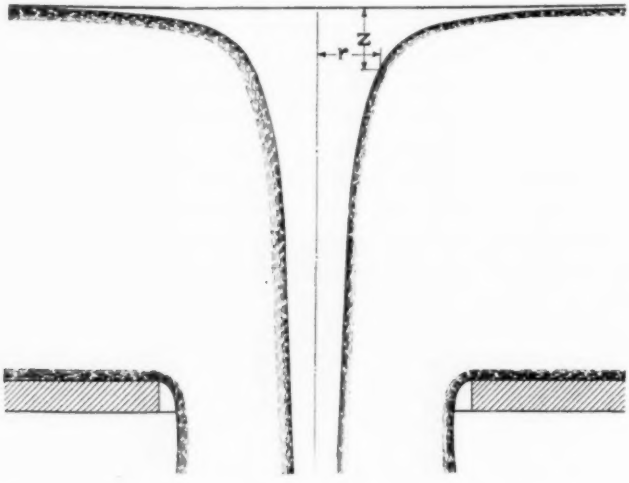


Fig. 2 Vortex Flow in Unobstructed Space of Revolution

water is set into an eddying condition without partaking of the general flow of the surrounding stream.

THE SPREADING TYPE OF DRAFT TUBE

14 The principle just explained suggests a useful method of regaining the kinetic energy of the whirling component of flow in the water discharged from a turbine runner or pump impeller. For example, if the flow is turned into an outward direction, away from the axis, the velocity of whirl will diminish in inverse proportion to the increasing radius, and the corresponding velocity head will diminish inversely as the square of the radius, so that it is merely necessary to lead the water a moderate distance away from the axis to obtain the conversion of a large proportion of the velocity head of

whirl into pressure head. This principle is used in the spreading type of draft tube.

15 The construction of this tube can be understood from Fig. 3, which shows a model of the draft tubes of the 30,000-hp. turbines for the United States Government Plant at Muscle Shoals, Alabama.

16 In utilizing this principle in the design of a turbine draft tube the meridian components of flow must not be lost sight of, and sufficiently gradual reduction in the meridian components must be provided, without sudden changes in direction. Both for the avoidance of rapid changes in velocity and direction of the meridian components, and also to avoid a surface of discontinuity and the

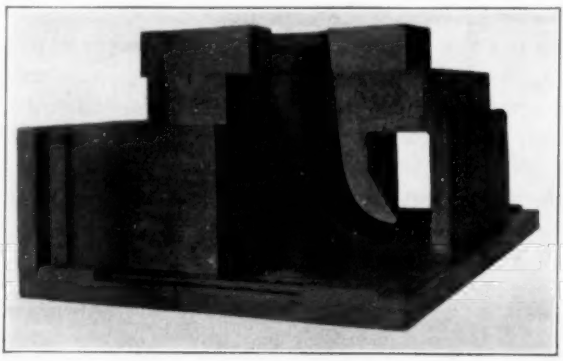


FIG. 3 MODEL OF SPREADING DRAFT TUBE OF 30,000-HP. TURBINES AT MUSCLE SHOALS PLANT

presence of a region of eddies, the value of including in the draft-tube design a central cone or core is clear. When a turbine draft tube is short, and when a long and awkward structure is not required to do it, it may be advisable to carry the central core up to the runner discharge, thus avoiding all flow close to the axis where the tendency to form a vortex or vortices is great. The writer also suggests the use of continuously curved inner and outer surfaces of revolution for the turbine water passage, thus avoiding all sudden changes in curvature. Fig. 4 shows in outline a turbine which follows these principles.

A NEW FORM OF TURBINE

17 In this new turbine form the entrance passage merges into the runner space and the latter merges into the draft tube, and the

flow passes into the draft tube by a continuous gradation, with gradual changes in the whirl about the turbine axis and without sudden variations of the secondary whirl occurring in the meridian plane, i.e., the plane of the figure.

18 If a ring of vanes such as a runner is interposed in the re-

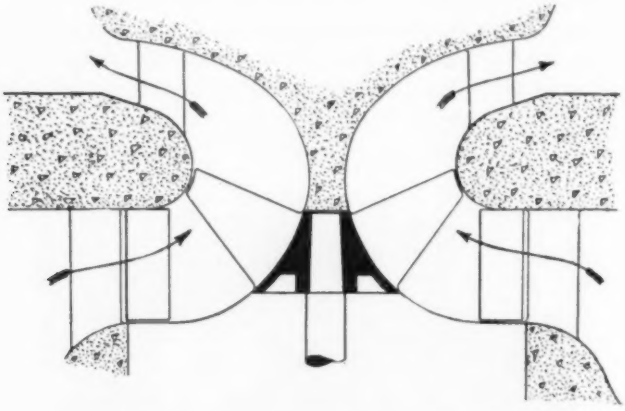


FIG. 4 A NEW FORM OF TURBINE

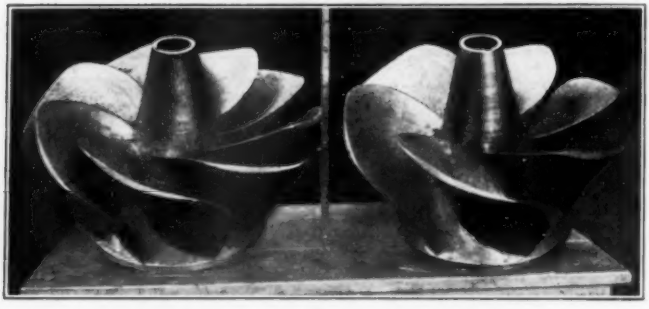


FIG. 5 NEW FORMS OF RUNNER SUITED TO NEW TYPES OF HIGH-SPEED TURBINE

volving mass of water, the torque exerted on the vanes by the water is equal to

$$\frac{W(r_1 c_{u1} - r_2 c_{u2})}{g}.$$

19 New forms of runner suited to the new types of high-speed turbine here considered are shown in Fig. 5.

DRAFT-TUBE AND RUNNER-VANE FRICTION LOSSES

20 Taking up some of the theoretical conditions presented by the modern forms of turbine, it will be recalled that in practically all of the time-honored turbine theories the design is based on the radial or axial direction of discharge from the runner; that is, for the conditions of best efficiency the absolute direction of out-flow from the runner is assumed to be in a meridian plane, a plane containing the axis. That this is not actually the best direction of discharge for high-speed runners has been known for several years, as found from pitot tube and direction-vane investigations in draft tubes.

21 With the use of the earlier forms of draft tube any direction of discharge having a whirl component introduced uncertainties and complications in the formulas, and the problem was not attractive. With the use of a draft tube capable of efficiently regaining the energy of whirl components as well as of meridian components of flow, however, it becomes possible to formulate some simple relations which it will be interesting to investigate. For one thing we shall no longer have to make the outflow loss from the runner dependent upon the direction of discharge, since we can use a draft tube which will handle the whirl components just as efficiently as the meridian components; the outflow loss from the runner can therefore be expressed as a function merely of the amount of the discharge velocity regardless of its direction. If the draft-tube efficiency is e_d and the coefficient of loss in the draft tube f_3, the portion of the velocity head of the water discharged from the runner which is lost or dissipated will be $f_3 c_2^2/2g$; c_2 being the absolute velocity of discharge from the runner and $f_3 = 1 - e_d$.

22 In runners of high specific speed another important loss is that due to the frictional resistance of the runner vanes, since very high relative velocities between the vanes and water are employed. In recent forms of high-speed turbines, such as just shown in Fig. 5, the rotational speed is high and the torque correspondingly low, so that the runner vanes have but little curvature and deflect the water only in small amounts. The loss of head due to the frictional resistance of the runner vanes can be expressed as $f_2 w_2^2/2g$, in which w_2 is the relative velocity with which the vanes move through the water, measured at their outflow edges. (For the reason just mentioned, however, the relative velocity is nearly the same over the whole vane.)

23 Instead of making any arbitrary assumption regarding the

direction of discharge upon which to base our turbine design, it will be useful to find the conditions which will give the minimum value for the sum of the above two losses — the outflow loss from the runner and the resistance loss in the runner. The problem may be stated as the determination of the conditions for maximum efficiency for a given specific speed or of maximum specific speed for a given efficiency.

24 Before proceeding with the problem itself, a distinction between two expressions for specific speed now in use should be explained. The usual formulation for specific speed used in turbine work is $N_s = N\sqrt{\text{hp.}}/H^{\frac{5}{4}}$. This form of specific speed is useful in turbine work because we most frequently have to fix the turbine speed when the horsepower output and the head have been decided upon. In centrifugal pump problems, however, we commonly define specific speed as $N_s = N\sqrt{Q}/H^{\frac{3}{4}}$, since the discharge and head are usually the known quantities. The two expressions are derived from the same dimensional relations and the second is just as applicable to turbines as to pumps, and the first as applicable to pumps as to turbines. Let us distinguish by calling the specific speed based on quantity $N_{sQ} = N\sqrt{Q}/H^{\frac{3}{4}}$. Although in adapting turbines of types already developed to new requirements of head and power the first expression is more convenient, the second has advantages when the problem is the development of a new design or type, since by its use we may design for a desired discharge under a definite head without introducing unnecessary uncertainty by having to assume an efficiency in advance. Moreover, if the efficiency should turn out to be materially different from that expected, the discharge and velocities will not be thrown so far out of agreement with the designed values if taken for a given N_{sQ} as they would be if taken for a given N_s. In stepping a small turbine up to a larger size the N_{sQ} will remain more nearly constant than the N_s.

25 In the problem in hand we will gain simplicity by employing N_{sQ}, and if we determine the conditions for maximum N_{sQ} for a given efficiency we shall at the same time have found those for the maximum N_s, since

$$N_s = \sqrt{\frac{62.4}{550}} \times N_{sQ}\sqrt{e}$$

and the conditions which will give the best efficiency for a given N_{sQ} will also give the best efficiency for a given N_s.

26 The writer might mention, in passing, his belief that the

knowledge of hydraulics has gained more from the study of dimensional relations such as the principles of similarity and specific speed than from any other method; and it may not be altogether fanciful to think of these relations as corresponding in a minor degree with the "relativity" movement in abstract philosophy.

27 To resume the problem of finding the conditions for the maximum value of the sum of the two losses mentioned, the "outflow triangle" of velocities is drawn as in Fig. 6(*a*) with the velocities marked to show the notation which is here used (the system which has now become standard in Europe).[1]

Here

c_2 = absolute discharge velocity from runner
w_2 = relative discharge velocity from runner
u_2 = velocity of runner
c_{m2} = meridian component of velocity of discharge
c_{u2} = whirl component of absolute velocity of discharge, which we may call the "absolute whirl," and
w_{u2} = whirl component of relative velocity of discharge which we may call the "relative whirl."

28 Since we are not concerned with any particular value of the head, we can adopt a "specific" value for each velocity, using the device of Thomann,[2] which is convenient in problems of this kind, and call

$$C_2 = \frac{c_2}{\sqrt{2gH}};\quad W_2 = \frac{w_2}{\sqrt{2gH}};\quad U_2 = \frac{u_2}{\sqrt{2gH}},\ \text{etc.}$$

and as we shall deal throughout with the outflow triangle, we can omit the subscripts (2).

29 The form of the triangle can easily be related to the specific speed by putting $U_2 = \pi D_2 N/60\sqrt{2gH}$ and $C_{m2} = Q/A\sqrt{2gH}$, in which A is the area of the turbine passage at the runner discharge measured normally to C_{m2}.

30 Then

$$N_{sQ} = \frac{60\sqrt{2gH}}{\pi D_2}\, U_2\, \frac{\sqrt{A C_{m2}\sqrt{2gH}}}{H^{3/4}} = \frac{60(2g)^{3/4}\sqrt{A}}{\pi D_2} U_2 \sqrt{C_{m2}}$$

or, dropping subscript (2), $k_{sQ}\, N_{sQ} = U\sqrt{C_m}$, in which

$$k_{sQ} = \text{a constant} = \frac{\pi\, D_2}{60\ (2g)^{3/4}\sqrt{A}}$$

[1] See Camerer, Wasserkraftmaschinen, 1914, pp. iv, vii and viii.
[2] R. Thomann, Die Wasserturbinen, 1908, pp. 11–12.



(*a*) (*b*)

Fig. 6 Outflow Triangle of Velocities

Velocity of runner vanes = $u_2 = (\pi D_2/60)N$. Meridian velocity component of water = $C_{m2} = Q/A$.
Specific speed based on quantity = $N_sQ = N\sqrt{Q}/H^{\frac{3}{4}} = (60\sqrt{K}/\pi D_2 H^{\frac{1}{4}})u_2\sqrt{C_{m2}}$

31 Before considering the problem with reference to the amounts of the vane loss and outflow loss, we can reach an interesting conclusion simply by supposing that the magnitude of the absolute and relative velocities C and W are kept constant while their directions are changed until the most advantageous shape of outflow triangle is attained. As shown in Fig. 6(b) various shapes of outflow triangle can be drawn with the same values of C and W, and therefore with the same losses of head. There will be one of these triangles which will give the highest specific speed, which we have just seen is proportional to $U\sqrt{C_m}$. To find the relation which will thus give the maximum N_{sQ} for a given loss, we can put

$$k_{sQ}\, N_{sQ} = U\sqrt{C_m} = (\sqrt{C^2 - C^2_m} + \sqrt{W^2 - C^2_m})\,\sqrt{C_m}$$

and considering C and W to be constant, we can differentiate $(k_{sQ}\, N_{sQ})$ with respect to C_m and equate to zero:

$$\frac{k_{sQ}\, dN_{sQ}}{dC_m} = (\sqrt{C^2 - C^2_m} + \sqrt{W^2 - C^2_m})\,\frac{1}{2\sqrt{C_m}} - \sqrt{C_m}\left(\frac{C_m}{\sqrt{C^2 - C^2_m}} + \frac{C_m}{\sqrt{W^2 - C^2_m}}\right) = 0.$$

From this we have, simplifying,

$$\sqrt{C^2 - C^2_m} + \sqrt{W^2 - C^2_m} = 2C^2_m\left(\frac{1}{\sqrt{C^2 - C^2_m}} + \frac{1}{\sqrt{W^2 - C^2_m}}\right)$$

and substituting C_u for $\sqrt{C^2 - C^2_m}$ and W_u for $\sqrt{W^2 - C^2_m}$,

$$C_u + W_u = 2C^2_m\left(\frac{1}{C_u} + \frac{1}{W_u}\right);\ \text{or}\ U = 2C^2_m\left(\frac{W_u + C_u}{C_u W_u}\right) = \frac{2C^2_m U}{C_u W_u}$$

which gives $C_m = \sqrt{C_u W_u/2}$ as the most advantangeous proportion. That is, the meridian velocity should be chosen as 0.707 multiplied by the mean proportional of the absolute whirl and relative whirl. In calculations applying to the runner as a whole, the velocities corresponding to an average point in the runner may be used, such as the center of area of a sector of the discharge space.

32 Fig. 7 enables us to visualize the meaning of this relation. The vertex of the outflow triangle should fall somewhere on the ellipse of major axis U and semi-minor axis =

$$0.707\sqrt{\frac{U}{2} \times \frac{U}{2}} = 0.354\ U$$

33 For example, if a runner has a vane inclination β_2 its probable point of best operation can be found by drawing W at this angle with the base line and completing the triangle with the vertex

at the point where the W line intersects the ellipse. According to this diagram, it would evidently never pay, from the standpoint of efficiency alone, to use a meridian velocity greater than 35.4 per cent of the circumferential velocity of the runner. Since the meridian velocity may be somewhat increased, however, before the efficiency is seriously impaired, it will probably be economical to use slightly higher values of C_m than are called for by this relation, in order to reduce the turbine dimensions and cost.

34 Now proceeding with the problem, and considering the amounts of the losses of head, we can derive some useful relations, as follows:

35 From the above expression for specific speed, we see that

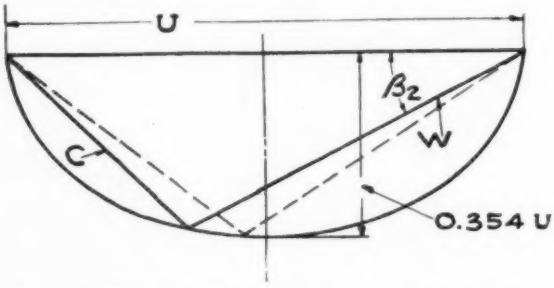


FIG. 7 DIAGRAM SHOWING PROBABLE POINT OF BEST OPERATION FOR RUNNER WITH VANE INCLINATION β_2

we can keep the length of base U of the outflow triangle and its altitude C_m constant, and can change its shape without changing the specific speed. By shifting the vertex parallel to the base, there must be some position which will make the sum of the outflow loss and vane resistance loss a minimum.

36 Calling the sum of these losses H_L, the loss of head expressed as a fraction of the effective head is

$$h_L = \frac{H_L}{H} = f_2 \frac{w_2^2}{2gH} + f_3 \frac{c_2^2}{2gH} = f_2 W^2 + f_3 C^2$$

37 Expressing h_L in terms of U, C_m and C_u,

$$\begin{aligned} h_L &= f_2 W^2{}_u + f_3 C^2{}_u + (f_2 + f_3) C^2{}_m \\ &= f_2 U^2 + f_2 C^2{}_u - 2 f_2 U C_u + f_3 C^2{}_u + (f_2 + f_3) C^2{}_m \\ &= f_2 U^2 + (f_2 + f_3) C^2{}_u - 2 f_2 U C_u + (f_2 + f_3) C^2{}_m \end{aligned}$$

38 For a given U and C_m we can find the value of C_u which will make h_L a minimum, neglecting the effect of any small variation

of f_2 due to a change in direction of W as being of a higher order of small quantities than differences in the losses themselves. Then

$$\frac{dh_L}{dC_u} = 2(f_2 + f_3)C_u - 2f_2U = 0$$

That is,

$$C_u = \frac{f_2}{f_2 + f_3}\,U, \text{ or } f_2(U - C_u) = f_3C_u$$

from which

$$f_2W_u = f_3C_u, \text{ or } \frac{C_u}{W_u} = \frac{f_2}{f_3}$$

39 This means that for the best efficiency the absolute whirl should be to the relative whirl as the coefficient of frictional loss is to the coefficient of outflow loss.

40 The above result shows for one thing that in a runner for which f_2 is large, as is the case in runners having a large amount of vane surface — particularly when this surface is increased by an outer band — the best condition of operation will be with comparatively low relative whirl and high absolute whirl, so that such runners will discharge the water at a greater angle of inclination to the meridian plane than runners having less vane surface and correspondingly lower f_2. The numerical value of f_2 can be calculated by figuring the loss of head in the runner buckets considered as rectangular channels. The purpose of this investigation is not so much to take up questions of design and calculation, however, as to establish some of the controlling relations, and we will give less attention to the numerical values of the coefficients than to the general conclusions which may result.

41 Having established the above relation between the relative whirl and absolute whirl at the runner discharge, we can now proceed, assuming that the said relation is to be complied with, to find the best relation between the meridian velocity C_m and the velocity of the runner U. Expressing the loss of head in terms of U and C_m in the expression

$$h_L = f_2U^2 + (f_2 + f_3)C^2_u - 2f_2UC_u + (f_2 + f_3)C^2_m$$

by expressing C_u in terms of U according to the result just obtained above, namely, $C_u = [f_2/(f_2 + f_3)]\,U$, we have

$$h_L = \frac{f_2f_3}{f_2 + f_3}\,U^2 + (f_2 + f_3)C^2_m$$

Substituting in this

$$U = k_{sQ}N_{sQ}/\sqrt{C_m}$$

we have

$$h_L = \frac{f_2 f_3}{f_2 + f_3} \times \frac{k^2_{sQ} N^2_{sQ}}{C_m} + (f_2 + f_3) C^2_m$$

To obtain the conditions for minimum h_L for a given N_{sQ} we will differentiate the last expression with respect to C_m, considering N_{sQ} a constant, and will equate the derivative to zero:

$$\frac{dh_L}{dC_m} = -\frac{f_2 f_3}{f_2 + f_3} \times \frac{k^2_{sQ} N^2_{sQ}}{C^2_m} + 2(f_2 + f_3) C_m = 0$$

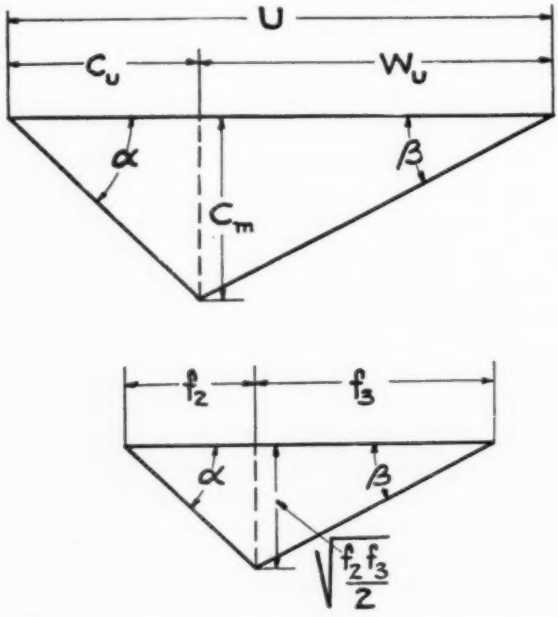


FIG. 8 RELATIONS OF VARIOUS OUTFLOW VELOCITIES FROM THE RUNNER TO EACH OTHER

from which, replacing $k_{sQ} N_{sQ} / \sqrt{C_m}$ by U, we have

$$2(f_2 + f_3) C_m = \frac{f_2 f_3}{f_2 + f_3} \times \frac{k^2_{sQ} N^2_{sQ}}{C^2_m} = \frac{f_2 f_3}{f_2 + f_3} \frac{U^2}{C_m}$$

From this we obtain

$$\frac{C_m}{U} = \frac{\sqrt{\frac{1}{2}(f_2 f_3)}}{(f_2 + f_3)}$$

42 The same result is called for by the relation $C_m = \sqrt{C_u W_u / 2}$, already given, if we insert

$$C_u = \frac{f_2}{f_2 + f_3} U \text{ and } W_u = \frac{f_3}{f_2 + f_3} U$$

For the minimum loss for a given specific speed the various outflow velocities from the runner should therefore be related to each other in the proportions shown at the top in Fig. 8, which is geometrically similar to the outflow diagram as indicated in the lower part of the same figure. That is, C_u, W_u, C_m, and U should be to each other respectively as $f_2, f_3, \sqrt{\frac{f_2 f_3}{2}}$ and $(f_2 + f_3)$.

From the last-mentioned figure it is seen that

$$\tan \beta_2 = \sqrt{\tfrac{1}{2}\frac{f_2}{f_3}} \text{ and } \tan \alpha_2 = \sqrt{\tfrac{1}{2}\frac{f_3}{f_2}}$$



FIG. 9 NEW TYPE OF HIGH-SPEED PUMP IMPELLER

43 In order to use advantageously small values of β_2, to secure runners of very high specific speed, it therefore becomes necessary to obtain low values of f_2, and thereby to avoid the necessity for high angles of whirl at the runner discharge. These small values of f_2 can be secured by reducing the exposed runner surface to a minimum. If this reduction of vane area is carried too far, however, another source of loss will be introduced due to permitting the passage of a considerable portion of the flow between the vanes without its acting effectively upon them.

44 The foregoing relations are as applicable to pumps as to

turbines, and furnish a helpful guide in proportioning the absolute whirl, the relative whirl and the meridian velocity component in designing both turbines and pumps of high specific speed. Fig. 9 shows a new type of high-speed pump impeller.

45 In looking forward to the future development of the turbine, it can be seen from the foregoing that theoretic considerations call for a conservative proportioning of the various velocities and angles of the turbine, and that there is no apparent means of greatly increasing specific speeds merely by using some new and radical proportion between the velocities. Efficiencies are already so high that no startling increase is possible. The prospect of further increases in specific speed therefore lies entirely in the use of higher velocity heads in comparison with the head on the plant, and an avoidance of serious impairment of the efficiency, by giving attention to the securing of low coefficients of loss in the runner and draft tube, and by observing the relations between the velocities deduced above.

POSSIBILITIES OF FURTHER INCREASE IN SPECIFIC SPEED

46 To get some idea of the possibilities of further increases of specific speed, it may be pointed out that if the best relation between the velocities is adhered to the losses here considered will vary as the four-thirds power of the specific speed N_{sQ}, the functional relation between efficiency and N_{sQ} being derived in the following manner,

$$h_L = \frac{f_2 f_3}{f_2 + f_3} U^2 + (f_2 + f_3) C^2_m; \text{ and from } \frac{C_m}{U} = \frac{\sqrt{\frac{f_2 f_3}{2}}}{f_2 + f_3},$$

$$h_L = 3(f_2 + f_3) C^2_m; \text{ and also, from } k_{sQ} N_{sQ} = U\sqrt{C_m} = \frac{f_2 + f_3}{\sqrt{\frac{f_2 f_3}{2}}} C_m^{3/2}$$

$$h_L = \frac{3}{2^{2/3}} \times \frac{(f_2 f_3)^{2/3}}{(f_2 + f_3)^{1/3}} k^{4/3}_{sQ} N^{4/3}_{sQ}, \text{ and}$$

$$\text{efficiency} = 1 - \frac{3}{2^{2/3}} \times \frac{(f_2 f_3)^{2/3}}{(f_2 + f_3)^{1/3}} k^{4/3}_{sQ} N^{4/3}_{sQ}$$

47 This considers the effect of only the two losses here treated and neglects certain other losses. These values of efficiency are therefore merely indications of the limiting values attainable.

48 The two losses mentioned account for nearly the whole loss in turbines of high specific speed. In low-speed machines two other losses assume importance — those due to leakage and disk friction.

These losses can be shown to be given approximately by the following formulas:[1]

49 Leakage loss, expressed as a fraction of the total quantity, is equal to

$$L_L = \frac{q_L}{Q} = \frac{201\,\phi^2}{N^2{}_{sQ}}$$

where ϕ has the usual significance; i.e.,

$$\phi = \frac{\pi\,D\,N}{60\sqrt{2gH}}$$

and disk-friction loss, expressed as a fraction of the total power, is equal to

$$L_D = \frac{l_d}{\text{hp.}} = \frac{35\,\phi^5}{N^2{}_s}$$

and for present purposes a still rougher approximation will be sufficient, giving

$$L_L = \frac{100}{N^2{}_{sQ}} \text{ and } L_D = \frac{6}{N^2{}_s}$$

50 When labyrinth seals are used for the runner, the above leakage loss, which is for ordinary seals, may be as roughly equal to $L_L = 50/N^2{}_{sQ}$ or even less. We shall find that these losses become negligible at moderate values of N_s and N_{sQ} and are of no importance in the high-speed field which we are particularly investigating.

TURBINE AND PUMP EFFICIENCIES AS RELATED TO SPECIFIC SPEED

51 In Fig. 10 are shown a series of curves of efficiency plotted against N_{sQ}. Curve A is the attainable efficiency computed from the theoretic relation derived above, using as values of the coefficients $f_2 = 0.04$ and $f_3 = 0.15$. To show the effect of altering the coefficients, Curve B has been plotted with $f_2 = 0.03$ and $f_3 = 0.20$. The effect of leakage and disk friction has been shown by deducting these losses from the efficiencies given by Curve A, and Curve C shows the result (the leakage loss being figured for turbines having labyrinth seals).

52 To show the general progress which has been made in recent years, and the continuing trend toward higher specific speeds, a

[1] For derivation see paper on The Specific Speed of Hydraulic Turbines by the writer, presented at December 28–30, 1908, meeting of Am. Assn. for Advancement of Science, and printed in *The Polytechnic* of Rensselaer Poly. Inst., 1913.

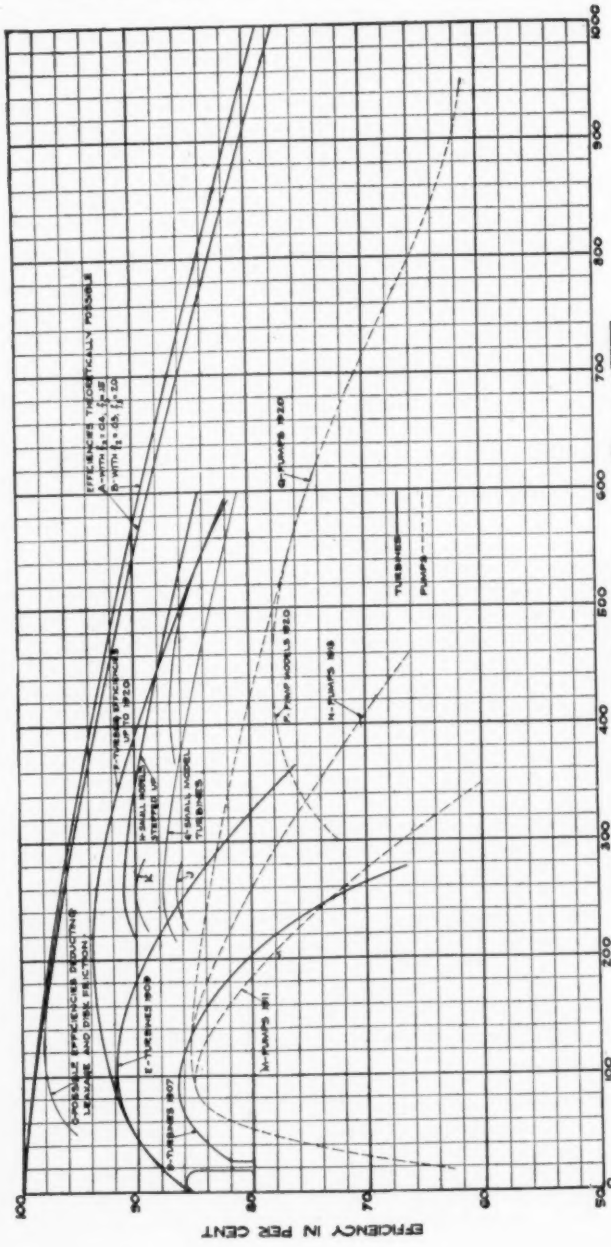


FIG. 10 CURVES OF TURBINE AND PUMP EFFICIENCIES PLOTTED AGAINST SPECIFIC SPEED BASED UPON QUANTITY N_Q

A — Calculated efficiencies based on loss in runner = $f_2 w_2^2/2g$ and discharge loss from runner = $f_3 c_2^2/2g$ with $f_2 = 0.04$, $f_3 = 0.15$, and neglecting other losses

B — Calculated efficiencies with $f_2 = 0.03$ and $f_3 = 0.20$

C — Calculated efficiencies deducting leakage loss ($= 50/N^2_s Q$) and disk-friction loss ($= 6/N^2$)

D — Turbine efficiencies attained in Europe up to 1907 (from paper by Graf and Thoma, *Zeit. Ver. Deutsch. Ing.*, June 29, 1907)

E — Turbine efficiencies attained in America and Europe up to 1909 [from discussion by L. F. Moody of paper by C. W. Larner in *Transactions*, American Society Civil Engineers, vol. lxvi, p. 306 (1910)]

F — Turbine efficiencies attained up to 1920 from tests so far reported

G — Efficiencies of small model turbines (16-in. diam.) in I. P. Morris Hydraulic Laboratory of Cramp Shipbuilding Co.

H — Efficiencies of Curve G stepped up to Holyoke size, based on comparison of Holyoke test (Curve K) and geometrically similar model (Curve J) of 16-in. diam. at I. P. Morris Laboratory

J — Test of small model of Cedars Rapids type turbine at I. P. Morris laboratory

K — Holyoke test of larger model of Cedars Rapids type turbine

L — Efficiencies from paper by Forrest Nagler, TRANSACTIONS, AMERICAN SOCIETY MECHANICAL ENGINEERS, vol. 41, p. 829 (1919)

M — Efficiencies of centrifugal pumps from Greene's Pumping Machinery, 1911

N — Efficiencies of centrifugal pumps up to 1913

P — Tests of 12-in. pump models of author's spiral type at I. P. Morris Laboratory up to 1920

Q — Pump efficiencies attained up to 1920

number of other curves have been plotted in the same figure, four of them being for pumps.

53 It is interesting to note the similarity in the form of curves *G* and *H* with *A* and *B*. As more development work is carried out in the high-specific-speed field, it may be expected that the efficiencies shown by Curves *F* and *Q*, for both turbines and pumps, will be materially exceeded and brought into closer relation to the theoretic values of Curves *A* and *B*, assuming that the coefficients used in computing these curves can be more closely approached.

54 The generally lower range of values of pumps as compared to turbines at all specific speeds is not entirely due to essential differences in action, but much of the difference the writer believes to be attributable to the generally smaller dimensions of pumps; although the pump suffers a definite disadvantage in its action from the fact that in the pump the impeller imparts velocity head to the

fluid and depends upon a stationary diffuser for the reconversion of this velocity head into pressure head; while in the turbine pressure head is converted into velocity head in the stationary casing and guides, and is absorbed in the runner, leaving only a small portion for reconversion in the draft tube.

55 Going back to the theoretic conditions for best efficiency or higher specific speed, let us see what is involved in the increases in specific speed shown by Curves *A* and *B*. These high specific speeds are imagined to be attained without changing the proportioning of the velocities relatively to each other, but by increasing the magnitudes of all of them, and thus requiring the use of higher velocity heads in comparison with the head on the plant. To see what this means in actual values, we can determine the manner in which the absolute velocity of discharge from the runner varies with respect to N_{sQ}. The derivative is as follows:

56 Referring to the proportions shown in the triangles of Fig. 8,

$$\frac{C_m}{C} = \frac{\sqrt{\frac{f_2 f_3}{2}}}{\sqrt{f_2^2 + \frac{f_2 f_3}{2}}}; \quad \frac{U}{C} = \frac{f_2 + f_3}{\sqrt{f_2^2 + \frac{f_2 f_3}{2}}}$$

We also have

$$k_{sQ} N_{sQ} = U\sqrt{C_m} = \frac{(f_2 + f_3)\left(\frac{f_2 f_3}{2}\right)^{1/4}}{\left(f_2^2 + \frac{f_2 f_3}{2}\right)^{3/4}} C^{3/2}$$

and hence

$$C^2 = \frac{f_2^2 + \frac{f_2 + f_3}{2}}{(f_2 + f_3)^{4/3}\left(\frac{f_2 f_3}{2}\right)^{1/3}} k_{sQ}^{4/3} N_{sQ}^{4/3}$$

or

$$C^2 = \frac{c_2^2}{2gH} = \frac{1 + \frac{f_3}{2f_2}}{\left(1 + \frac{f_3}{f_2}\right)^{4/3}\left(\frac{f_3}{2f_2}\right)^{1/3}} k_{sQ}^{4/3} N_{sQ}^{4/3}$$

57 From the last equation values of C^2, which represents the ratio of the velocity head of the water at discharge from the runner to the head on the turbine, can be computed for various values of the specific speed. As previously noted, the term k_{sQ} in the formula for C^2 is a coefficient having the value

$$k_{sQ} = \frac{\pi D_2}{60\,(2g)^{3/4}\sqrt{A}}$$

In this expression D_2 should be taken as the mean diameter of discharge of the runner and A the cross-sectional area of the space into which the runner discharges, measured in a section perpendicular to the meridian velocity. In a runner in which the direction of the meridian velocity of discharge is at a moderate inclination to the axis and having vanes extending nearly to the axis, D_2 may be taken

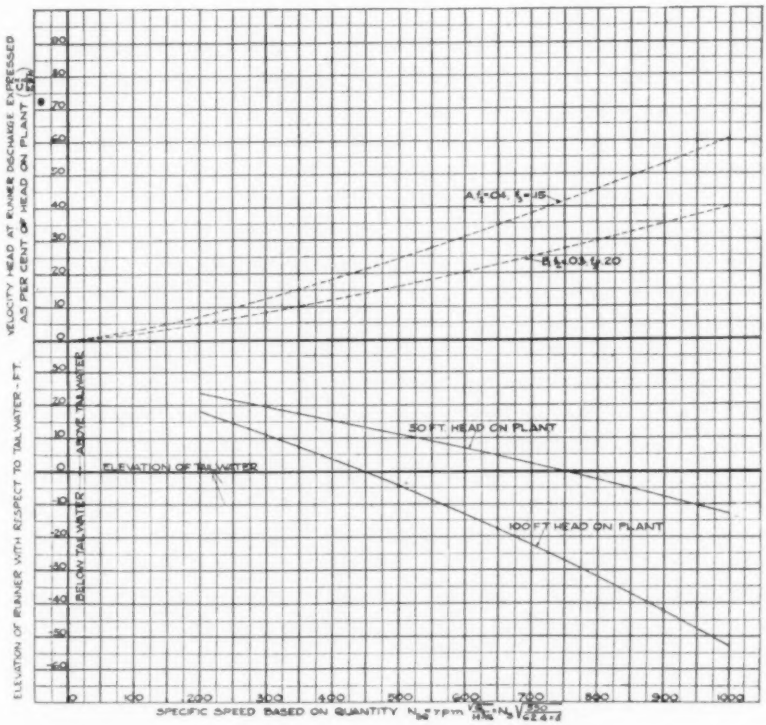


Fig. 11 Curve Showing Restriction Placed upon Specific Speed by Location of Turbine with Respect to Tailwater

as approximately equal to $\frac{2}{3}$ D, where D is the diameter of the outflow space. This gives us

$$k_{sQ} = \frac{\sqrt{\pi}}{45\ (2g)^{3/4}} = 0.00174$$

58 The computation has been carried out for the two sets of values of f_2 and f_3 which were used in plotting Curves A and B in Fig. 10. Fig. 11 (upper diagram) shows the corresponding values of

C^2, the curves in this figure being marked A and B to correspond to Fig. 10.

59 The ordinates of these curves represent the velocity head corresponding to the absolute velocity of discharge from the runner, expressed as a percentage of the head on the turbine. With the low values of the coefficients f_2 and f_3 which have been used in computing these curves, the possibility is shown of developing much higher specific speeds than have yet been attained, and it is also seen that this can be done without the necessity of employing extremely high velocity heads. It may be found from later experience that somewhat higher velocity heads may be employed without serious loss in efficiency, in order to reduce the turbine dimensions. The exact numerical values of coefficients f_2 and f_3 which can be secured in the future remain for further investigation to show.

60 Whatever the values of the coefficients secured by a particular turbine design the relation indicated by the general form of the curves of the last figure will always apply, i.e., increases in specific speed to extremely high values will inevitably involve, sooner or later, the use of higher velocity heads at the runner discharge. In turbines arranged according to present practice the extent to which the increase in specific speed can be carried will be limited by the height of the turbine runner above the surface of the tailwater.

61 In order that the absolute pressure at the runner discharge should be kept at a sufficient margin above the point at which the water will vaporize, and to provide the necessary head for the retardation of flow during quick gate closures, the sum of the static elevation of the runner above the tailwater surface and the regained velocity head should not approach unduly close to the barometric limit of 34 ft., and in the following examples the limit will be set at 27 ft. With such a limiting value, if a runner should be placed, say, 17 ft. above tailwater, there would remain only 10 ft. as the limit of the velocity head which can be employed, making the necessary allowance, however, for draft-tube losses, distribution of velocity, and increase in velocity during full-gate operation.

62 In order to show in a general way the restriction placed upon specific speed by the location of the turbine with respect to tailwater, the curves shown in the lower part of Fig. 11 have been drawn, using Curve A in the upper portion of this figure as a basis, but adding 30 per cent to the velocity heads called for by this curve in order to provide a margin to cover certain factors, including the unequal distribution of velocities at the runner discharge and the

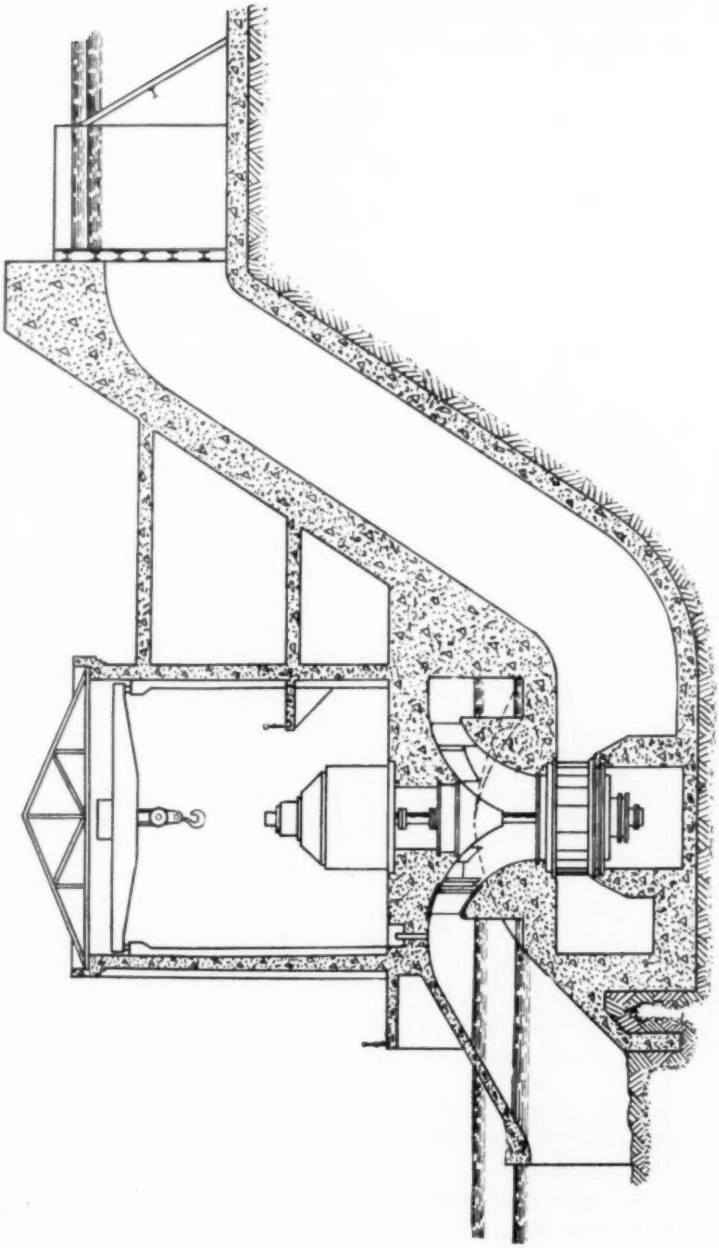


FIG. 12 NEW TYPE OF POWER PLANT

variation in the absolute velocity from the normal point of operation up to full gate, the necessary margin being reduced, however, by the fact that velocity head is not completely regained by the draft tube.

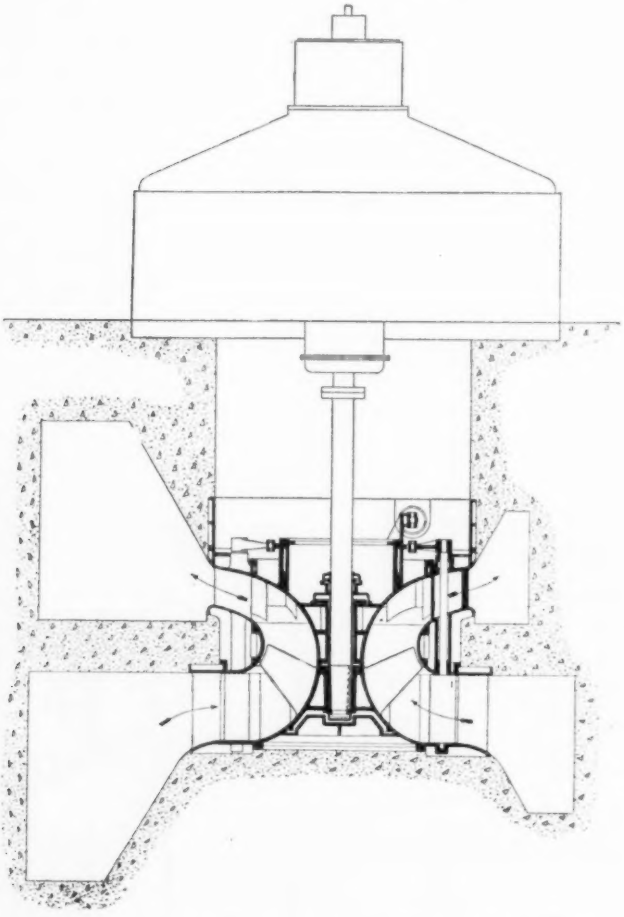


Fig. 13 Inverted Turbine of the Wicket-Gate Type

Two curves are shown in this figure, for plants operating respectively under heads of 50 ft. and 100 ft.

63 It will be noted that even with the low outflow losses called for with the types of turbines considered, a point is soon reached where the turbine must be set at the tailwater elevation or below it.

If the turbine should be applied to heads higher than 100 ft., this point would occur at still lower specific speeds.

POWER PLANTS WITH INVERTED TURBINES AND DRAFT TUBES DIRECTED UPWARD FROM RUNNERS

64 If we should adhere to the usual practice in the design of power plants, the future extension of the available range of specific

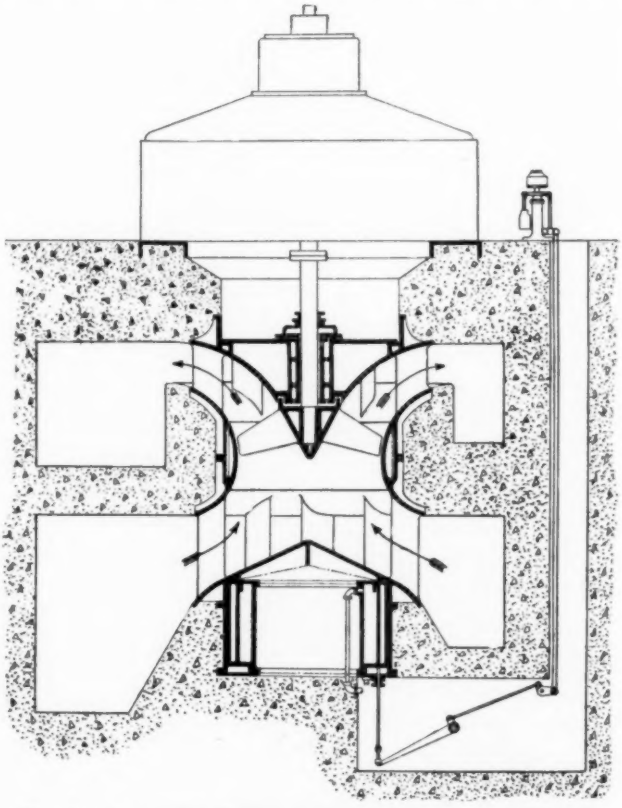


Fig. 14 INVERTED TURBINE OF THE PLUNGER-GATE TYPE

speeds for turbines would be narrowly restricted. The following procedure has, however, been proposed by Mr. H. Birchard Taylor, namely, that we remove this restriction on future progress and abandon the universal practice of placing the turbine runner at a considerable height above tailwater. Mr. Taylor proposes, if necessary,

to locate the runner considerably below the normal tailwater elevation. The objection immediately presents itself that even if the runner is placed above normal tailwater it will be submerged and inaccessible during times of high water. This difficulty is overcome by the very simple but somewhat radical procedure of turning the turbine upside down and directing the draft tube upward from the runner, so that it will discharge over a crest located slightly above the highest tailwater elevation to be encountered in a particular installation. Fig. 12 shows a power plant as it would be built in accordance with this idea, and Figs. 13 and 14 two new forms of inverted-type turbine which could be used in such a plant.

65 In Fig. 12 the turbine is arranged with wicket gates or movable guide vanes, operating from below in a pit provided for the purpose. The draft tube is of the spreading type, the outer barrel of the tube being used to form a circular crest to exclude the tailwater when access is desired to the interior of the turbine. The procedure to follow would be to close the head gates and to pump out the water contained by the draft tube and turbine passages by the use of low-head centrifugal pumps or hydraulic ejectors.

66 Without going into all of the new features opened by this arrangement, it may be mentioned that the turbine casing, which is under a considerable amount of pressure, is located at a low point in the power-house substructure where there is a large mass of concrete above it to counteract the upward pressure of the water. An excellent design of draft tube can be used without requiring any increased depth of excavation, and the runner is naturally located at a low point in the substructure.

67 The arrangements shown in Figs. 13 and 14 can be readily understood from the drawings. In the turbine in Fig. 13 wicket gates are used with the guide-vane stems carried up through, or in line with, the draft-tube stay vanes which are used to support the superimposed structure, and "outside" operating mechanism is used, arranged in the usual manner in the turbine pit. In Fig. 14 a plunger gate is used instead of the usual wicket gates, this plunger being operated by hydraulic pressure.

68 In conclusion, the writer hopes that in pointing out some of the possibilities for the future development of the turbine, the suggestions presented in the foregoing pages may prove helpful to other engineers working in this field.

No. 1825

THE RISING IMPORTANCE OF OIL-INJECTION TYPE OF INTERNAL-COMBUSTION ENGINE

By Charles E. Lucke, New York, N. Y.
Member of the Society

THE first successful commercial machines of the internal-combustion class were gas-burning engines. While they have now sunk into a condition of more or less commercial insignificance, they have left behind a useful influence in that they have taught certain lessons that are of value in solving the problems of liquid-fuel adaptation.

2 The first lesson taught by the gas-burning engine is, the higher the efficiency the higher must the compression be. That is as it should be on thermodynamic grounds, and experience has amply demonstrated the validity of the theory. The second lesson is, that in addition to high compression, high efficiency is obtainable only if combustion is carried out in a correct and proper manner as to timing and rate. The gas-burning engine has also demonstrated that to get the maximum results in both power and efficiency it is equally necessary that the fuel be intimately and homogeneously mixed with the air throughout its entire mass, and that the cylinder be fully charged with that kind of mixture.

3 In addition to these principles of combustion for transforming high percentages of heat into work, the building of gas-burning engines has established many basic principles in the structural problem. To make cylinders, pistons and heads that will not crack is not as easy as it would seem, but taking the experience of the world at large, it can be said that reliable means of avoiding cracks have been devised.

4 In the adaptation of liquid fuel there are certain special problems that have to be faced that did not exist with the gaseous-fuel internal-combustion engine. The two principal classes of prob-

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Condensed from a paper presented at a joint meeting of the Boston and Worcester Sections of The American Society of Mechanical Engineers, the Boston Section of the American Institute of Electrical Engineers, and the Harvard Engineering Society, Cambridge, Mass., March 23, 1921.

lems are in the fuel itself and in the special type of service to be met. It very early appeared there could be no such thing as a universal liquid-fuel engine equally good for gasoline, kerosene or fuel oil, or equally good for boats or automobiles or aircraft. There might very well be an automobile engine or a motorship engine, or a tractor engine, or a railroad engine, or a stationary electric-lighting-set engine, but each must be different.

5 It is this adaptation that occupies most of the period of development. To study the fuel phase of liquid-fuel adaptation, fuels must be divided into the two classes that are now found commercially but which division originally was not so clear. The first class includes fuels that are sufficiently volatile to make a more or less homogeneous and gaseous mixture with air by passing through so simple a device as a carburetor, which is similar to the older air-gas mixing valve of all gas-burning engines. The second may be termed the non-volatile class, and it includes anything that cannot be used in such a carburetor with or without heat, but which requires a device that must be built into part of the engine structure rather than an attachment to what would otherwise be a gas engine, thus initiating the injection oil engine.

6 The really difficult problem of the gasoline engine appears only when it is realized that the fuel available is no longer volatile enough to make the desirable homogeneous mixture, but not yet bad enough to require an injection engine. Before getting down to the problem of the injection engine proper, however, it is desirable to analyze some of the difficulties encountered in adapting the gas-burning engine and its principles of good utilization to light, and then to heavier, gasoline.

DIFFICULTIES ENCOUNTERED IN ADAPTING THE GAS-BURNING ENGINE TO LIGHT AND THEN TO HEAVIER GASOLINES

7 The first principle of maximum compression cannot be carried as far as is desirable because the ignition temperature of these gasoline mixtures is lower than that of the gases forming the bulk of the fuels for the more efficient gas-burning engine. Furthermore, the temperature of the mixture before compression is no longer under the complete control it used to be with cold gas, and the temperature at the point of ignition or when compression ends is as much a function of the temperature before compression begins as it is of the amount of pressure rise.

8 The conclusions reached in gas-burning engine practice with regard to mixture quality, proportionality, homogeneity, intimacy, are all verified with gasoline. In proportion as that kind of mixture is attained with the volatile liquid fuel, so is it possible to attain some fair measure of the promise of efficiency, but the limit of attainment and the realization of it both fall off as the volatility falls off, and with the gasoline we are now using approximately only one half vaporizes in the intake passages.

9 The other half that will not vaporize is carried along in three different states: (*a*) as a film on the walls, such as rain will form on a windowpane, (*b*) as a fog that floats, and (*c*) as a rain that is falling or driven by the air currents. A fog turns into rain, the rain makes films and the action cannot go far before the fuel is all wall film. No amount of ingenuity with devices such as screens, baffles and paddles can prevail, because these cannot overcome the laws of liquid flow and vapor pressure that are operating. This is a problem of serious importance, because if the unvaporized liquid fuel gets into the cylinder and strikes a hot spot, such as the piston head, it will vaporize there locally, and will form on the piston head a "pancake" of vapor that will have displaced the air and not be mixed with it. This when combustion takes place above it, is simply heated to decomposition temperature, producing carbon, and though actually not burning itself, it fouls up the engine and interferes with its operation with a loss of fuel.

10 If the unvaporized fuel entering the cylinder strikes on a cold cylinder wall, it will run down past the piston into the lubricating oil. This unvaporized fuel is mainly kerosene and the lubricating oil is also a petroleum product. They are mutually soluble and as a consequence the viscosity disappears. The mixture is no longer a lubricant. It runs down into the crankcase and destroys the lubrication of the main bearings and the crankpins, as well as the piston pins. How can these things be prevented from happening? Unless they are prevented, the engine is no longer commercial.

11 Two courses of action are open. The first is to heat the mixture as it leaves the carburetor, and thereby raise the vapor pressure to a point where in a 15 to 1 proportion there will be a vaporized-fuel and air mixture at the minimum possible temperature and a pressure of one atmosphere. It may be said, therefore, that a moderate amount of heating is permissible, and possibly a sufficient amount to completely dry the mixture if the gasoline is not too heavy, but complete heating for a kerosene mixture is not. Gain

will be realized, but also a loss, and the loss will overbalance the gain, and the practice must be abandoned.

12 The next mode of attack is to try to handle the mixture with some of the fuel as a liquid. To handle a wet mixture means really, in the modern multi-cylinder engine, to distribute the stream of liquid as it runs along on the inside of the pipes, to four, six, or more branches, giving to each the same amount of liquid in order that all cylinders may work the same, assuming the liquid will be vaporized or sprayed as it enters each cylinder. To accomplish this it is necessary to know how the liquid moves. Imagine the liquid coming up the side walls of the riser and approaching a bend as in Fig. 1. How would it turn? Observations in glass show that the liquid forms a very substantial lump at *A*, just beyond the turn,

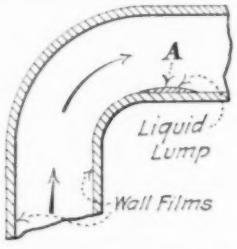


FIG. 1 GASOLINE-KEROSENE CARBURETOR ENGINE, LIQUID ON MANIFOLD WALLS

and on the inside of the bend. Films collect at the outside of the bend, but the velocity of the air-vapor mixture is so great at that point as to drive the liquid film around the bend to the point of least velocity. It is really the shape of the stream of air that the plug of liquid reveals.

13 Therefore, if the gasoline is not very heavy, then a moderate heating of the mixture — not too much — with some form of hot spot is the right thing so as to avoid preignition, and with this moderate heating a manifold to take care of the distribution of the rest of the liquid.

REMEDIES PROPOSED

14 The net result of all this is that the gasoline carburetor engine is approaching a crisis in its history that is going to force the use of radical remedies. The remedies now being considered are as follows:

15 First, the elimination of the manifold entirely. This will

take away the distribution problem and will permit the delivery of the liquid as a liquid with its air into the cylinder directly. No particular nozzle spraying is needed and not much vaporizing, because by properly forming the inlet valve and its passages a combustible foglike mixture will be formed as the charge enters the cylinder. This method has proved to be successful, and it is now the standard in use for all farm and most tractor engines, several hundred thousand of which are made every year, burning kerosene without any mixture heating whatever beyond what is incidental to suction.

16 Lubricating-oil contamination still is troublesome, and all such engines suffer from it to a greater or less extent. To minimize this the cylinder lubrication must be kept separate from that of the main bearings and crankpins. With admission of the mixture, however poorly vaporized, directly to the cylinder, good mixtures can be made without reduction of compression, but it is actually increased by adding water. This allows the water to enter the cylinder as a spray, just as the kerosene does, and by reason of its thermal, and to some extent its chemical, action, the compression can be raised so that some tractor engines and many farm engines have as high as 90 lb. compression.

17 The second remedy is to change the volatility of the gasoline by mixture with another and more volatile fuel. By properly selecting the things to be added to the gasoline, it is possible to not only improve the volatility, but at the same time raise the ignition point. Experiments with a benzol-alcohol-kerosene mixture at Columbia University have yielded truly wonderful results judged by the possibilities of the future. It is amazing to what extent compression can be raised on a charge of gasoline ready to preignite normally, with a minor amount of alcohol added. This is also true with benzol. Such mixtures carried the German aircraft through the war.

ABANDONMENT OF THE CARBURETOR ENGINE AS A MEANS OF SOLVING THE PRESENT MOTOR-FUEL PROBLEM

18 The third remedy — and this is the radical thing — is to abandon the mixture engine entirely and take up the injection engine. The abandonment of the carburetor engine, or the complete premixture engine, is a thing that would solve the present fuel problem as we hear of it, and that fuel problem is a real problem. America is today facing a situation in its liquid-fuel supply that is one of the

most serious things that has happened industrially. This country has reached the point where imports exceed exports. That means a real shortage. It is due to the motor-car and allied demand. Its direct effect is increased prices for those volatile constituents that are in greatest demand, but without a corresponding increase in price for such residuals as are not in demand. A switchover from the carburetor engine, which requires the light distillates, to an injection engine — independent of the distillate and of a kind which is independent of volatility, operating with anything having proper fluidity — would mean that the automobile industry would be revolutionized. Such a step cannot be taken suddenly. It is a difficult job, but it is possible to explain the difficulties and to state the progress that is being made in solving them. Not only is this a matter of interest to the engineer, but it is a matter of national importance.

19 In case of the successful development of a suitable injection engine the problem of gasoline shortage disappears, because the injection engine can handle any petroleum distillate, any coal-tar product, any alcohol or similar fuel, subject to the one condition that it shall be of proper fluidity to pass the pump valves and spray finely at the spray orifice. If a fuel is not naturally of high fluidity, there is not one that cannot be made of proper fluidity by adequate heating. The temperature of heating, however, must not be carried so far as to cause a decomposition with carbonization and cracking.

INJECTION AND CARBURETOR ENGINES COMPARED

20 The injection engine, compared with the carburetor engine, not only makes engines independent of the grade of fuel (there is only one requirement besides its viscosity — cleanliness), but it presupposes that there will be under compression only air, and that only after compression shall the fuel be injected. This means that by properly choosing the time of injection the compression may be as high as pleases the designer. There is no longer any limit of temperature of ignition, because there is nothing to ignite. As a consequence the injection engine has higher possibilities of efficiency, and those possibilities are attainable, and attained.

21 Another difference found between the injection and the carburetor engine is that due to the difficulty in making the fuel reach all of the air. With the gas-burning engine and with the carburetor type of volatile-liquid-fuel engine, a mixture is made externally and every part is actively combustible, so that there will be the maxi-

mum possible work per cylinder charge for a given compression and shape of combustion line. With the injection engine there are certain real advantages, as pointed out, but if after compression of the air charge the fuel is quickly thrown in, it will be difficult to reach all of the air in an unfavorably shaped combustion chamber from a single point of injection.

22 There must be a means of spraying the oil into a charge of dense air — air of a density up to 30 or 40 atmospheres — means of arranging to get the injected fuel in contact with as much of the air as possible, and means of preventing the delivery at any point of any considerable amount of fuel that cannot reach air, because in that case carbon will be formed and smoke produced which will choke up the engine in time.

THE TWO CLASSES OF INJECTION ENGINES

23 The means of carrying out the operations that are peculiar to the injection engine are divisible functionally into two classes. The air must always be compressed. It may be compressed to ignition temperature and higher, so that the fuel as injected into it ignites immediately and burns as fast as it gets in, the rate of combustion being the rate of injection and controlled by mechanical means. On the other hand, it may be compressed not to ignition but to something less than ignition temperature, and then the fuel injected suddenly to form an explosive mixture which will burn as nearly instantaneously as may be. This gives us two classes of injection engines. The first, the correct operation of which is shown in the indicator card of Fig. 2 in full lines, is explosive in type, and if the combustion is imperfectly carried out there may result explosive shocks as shown at A, or slow burning as at B in dotted lines, the other, shown in Fig. 3, carries the air to a higher compression pressure, so as to have it not only as hot as the ignition temperature but somewhat hotter. This will give a non-explosive combustion at substantially constant pressure according to the full line, but if combustion be carried out imperfectly, or deranged as to timing of injection or combustion, it may produce explosive shocks as at A or slow burning as at B according to the dotted lines.

24 Each of these two classes is subject to certain derangements or diseases peculiar to a given mechanism, but comparing the two properly adjusted, how do they stand with reference to each other? Is there any great choice, any reason why the advocate

of one should call the advocate of the other wrong? Not at all. Each is justifiable on the grounds of efficiency, power and practicability, so that the real problem boils down to one of mechanical questions of relative cost, reliability, foolproofness and adaptability to service conditions. The two are related in this simple manner as to efficiency. If the compression of the first with explosive-type combustion is about half the compression of the other burning the fuel at substantially constant pressure, their efficiencies are substantially the same. In the former case, if an explosive mixture is to be made, the compression must be kept X degrees below the ignition value in order to keep it under control, and in the other case Y degrees above ignition temperature to insure prompt ignition after injection.

25 Assuming the exponent n in the equation $PV^n = C$ to

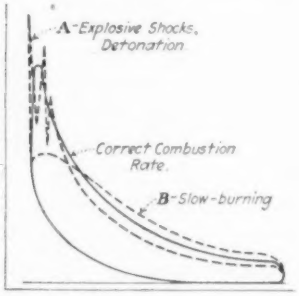


Fig. 2

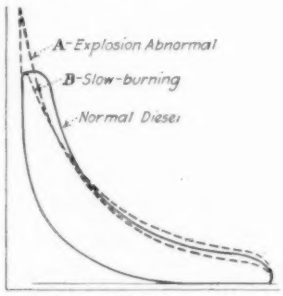


Fig. 3

FIG. 2 INJECTION OIL ENGINE — EXPLOSIVE COMBUSTION
FIG. 3 INJECTION OIL ENGINE — NON-EXPLOSIVE COMBUSTION

have a value of 1.4, it appears that 150 lb. compression will produce the ignition temperature of kerosene (998 deg. fahr.) if the initial temperature is a little less than 250 deg. fahr., and if an explosive mixture is to be formed and not preignited, a margin of 100 deg. below ignition will be attained with an initial temperature of something under 200 deg. and a margin of 200 deg. with about 175 deg. fahr. initial. For fuel oil having an ignition temperature of 1070 deg. fahr. the same conditions will be brought about by the same compression when the initial temperatures are 300 deg. fahr., 250 deg. and 200 deg. respectively.

26 On the other hand, if the air is to have a safe margin of temperature over the ignition value before compression, higher compression or higher initial temperatures are necessary. For solid-

injection sprays it is generally assumed that 200 deg. margin is safe and for air spraying 400 deg. margin. For the latter case a compression of 450 lb. is pretty generally adopted, and this will be secured with a little over 250 deg. initial with kerosene and a little less than 250 deg. with fuel oil. Solid injection ignition may be produced with equal reliability with less compression, or with lower initial temperature, or both.

METHODS OF SPRAYING FUEL EMPLOYED IN INJECTION ENGINES

27 The first mechanical problem in connection with this injection engine is that of making the spray, and one might say, in a way, that the building of the engine begins with the forming of a chamber around a spray. The simplest way of making a spray nozzle, introduced by the first successful commercial engine which was

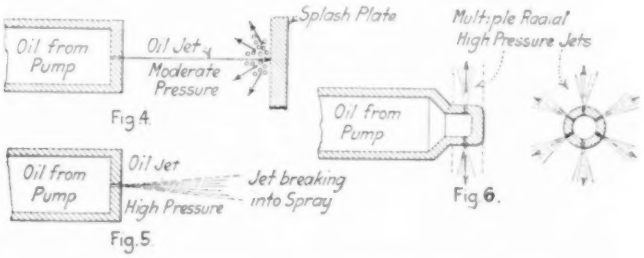


FIG. 4 HORNSBY JET-SPLASH SPRAYING DEVICE
FIG. 5 HIGH-PRESSURE JET BREAKING INTO SPRAY
FIG. 6 MULTIPLE RADIAL HIGH-PRESSURE JETS BREAKING INTO SPRAY

brought here from England — the Hornsby — consists in drilling a hole in a plate. Through this a jet is projected which strikes the wall, a spray being formed by splashing (Fig. 4). If the hole is reduced in size, or if supplied with oil under much higher pressure, the oil will move proportionately more slowly on the sides as compared with the center, and finally the entire jet will expand into a fine mist (Fig. 5).

28 The essential characteristic of such a spray is strong penetration power. What is necessary, however, is some means of spreading. This can be secured by multiple holes (Fig. 6). It can also be secured by using slots which are normally closed but which are opened by the oil pressure deflecting the metal. Spread can be secured also from a single spray orifice by giving the oil back of the orifice a rotary motion just as in the mechanical atomizer oil burner

as developed for the Navy (Fig. 7). Here the oil grooves are arranged to deliver tangentially into a small whirl chamber A, so that the oil in the chamber will have a rotary motion, as also will the oil issuing from the end of the orifice B at the outlet from the chamber. If the orifice B has an area many times greater than the area through these grooves, there will be no residual pressure in the chamber A, and in this case with sharp edges oil spray will issue in a hollow cone form, due to a pure centrifugal whirl. The spray will have a good uniform spread with little or no penetration. If, however, orifice B be narrowed down so that it is smaller than the grooves in area,

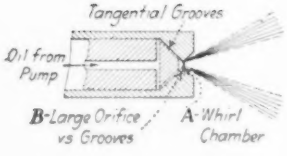


Fig. 7 Single-Orifice Whirl-Chamber Spray Nozzle

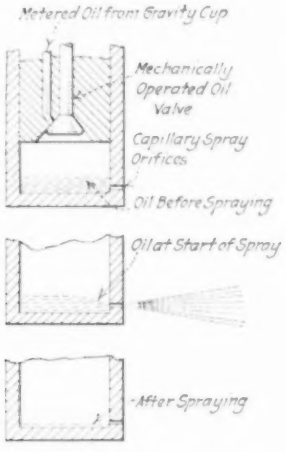


Fig. 8 Hvid Air-Spraying Cup

there will be residual pressure in the whirl chamber A, producing axial velocity also, and the cone spray becomes narrower and solid. The spray will then have less spread and more penetration.

29 In addition to this method of solid-injection spraying there is the air-spray system. This in its simplest form (Fig. 8) consists of a cup with liquid fuel in the bottom, compressed air above the fuel, and a small hole in the side at or below the oil level. As the air escapes through the hole there is first a depression of the liquid right at the point and the liquid is carried to the orifice by the air flow across its surface and blown out, being sprayed by the higher velocity of the air. Such a spray is fine and has good penetration, but not much spread.

30 A modification giving somewhat better control is shown in

Fig. 9, which has a depression in the passageway at A, in which the fuel is deposited as a pool. A deflector over the oil directs the air down and across its surface. The air in motion will tear the liquid off from the surface and this may be delivered to the cylinder in a narrow cone spray through a contracted orifice A, or through multiple orifices as at B, to get an adequate spread. This form is the so-called "open air spray" of the Diesel engine.

31 Next there is the so-called "closed air-spray valve," in which a valve seats on the outlet of a passage as in Fig. 10, and which when lifted allows the air to flow. The oil is delivered by the pump into the passage and spreads out on plates usually provided with holes, grooves, or slots so as to offer a large amount of surface to be

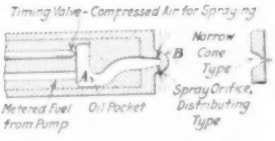


Fig. 9 Air-Spray Valve, Open Type

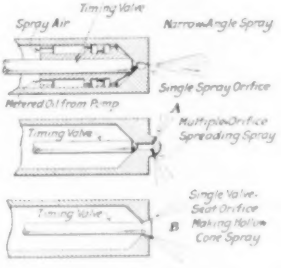


Fig. 10 Air-Spray Valve, Closed Type

wetted by the oil, which can be blown off gradually and delivered as a spray through a single or multiple orifice A. Such a spray has a narrow angle, strong penetration, and little spread except as may result from impact and rebounding. This is the most common air spray of the Diesel engine. To secure more spread directly, the valve can be reversed in seating as in B.

32 For any given form of spray, air or solid, there must be a suitable combustion-chamber form, or for any given combustion-chamber form there must be selected a spray of suitable shape or energy to best reach distant air, with always the possibility of setting up turbulence or internal air currents as a corrective means.

DIESEL ENGINES AND THEIR LIMITATIONS

33 The first class of engines to be noted under the injection type is the one that is most successfully used commercially — the

air-injection Diesel engine — and which normally has about 450 lb. compression — more than is sufficient to ignite an ordinary oil, enough more to be safely above ignition temperature all the time. Such engines are normally rated at 70 lb. brake mean effective pressure but are capable of producing over 100 lb. if the metal can stand the intense heating. A fuel consumption of from 0.4 to 0.45 lb. per b.hp. per hr. is standard. They are built in all except very small sizes with cylinders up to about 36 in. diameter, depending on speed and mean pressure. The maximum size is limited by the same internal heating conditions with tendency to crack the metal as obtain for large gas engines. The air-compression Diesel engine is in successful use, as is well known, for both stationary and marine purposes, and is the standard oil engine in use for large ships.

34 Certain limitations of the air-injection Diesel engine make it unsuitable as a substitute for the gasoline engine. It cannot be made to work with cylinders of too small a size without abnormally high compression, because of the cooling conditions that exist during compression, and in the smallest practical size it is too expensive and too complicated. The control of the air spray is peculiarly delicate. The air for it must be provided by an attached air compressor, and it requires a pressure of never less than 600 and often 1300 to 1500 lb. per sq. in. Such a compressor small enough for the purposes of an automobile is a mechanical absurdity. The particular field to which the air-injection Diesel is not at all adapted, therefore, is that of the small-cylinder high-speed engine, and that, in the internal-combustion market, is the biggest field of all.

DESIRABLE FEATURES OF SEMI-DIESEL ENGINES

35 To approach the problem of the small injection engine, what is available as a starting point? The nearest thing is a type of engine that has been on the market for some time and which is commonly known as the semi-Diesel. These have a feature particularly attractive in the small-engine field — that of operating with so-called "solid-injection or airless spray" that eliminates the air compressor and the delicacy of adjustment of an air-spray valve system. They are simple, but do not operate nearly so well as the air-injection Diesel engine. One feature of this class is hot metal, which plays in these engines the function of more or less vaporizing the fuel and also and mainly that of ignition. This hot metal, when

an external wall, is always a fire risk. On a ship it may be serious and in many buildings it is prohibitive. A hot-metal combustion chamber is practically an auxiliary pressure-enclosing wall and thereby constitutes an element of some danger of breakage. The temperature of the hot metal is difficult to control within proper limits, and sometimes impossible.

36 These semi-Diesel engines of hot-bulb, plate or tube pattern cannot be described in detail because of lack of space. They differ from each other mainly in the hot-metal form, or location, and the combustion-chamber shape with reference to oil injection. An unjacketed cap *A* (Fig. 11), more or less hemispherical, closing a

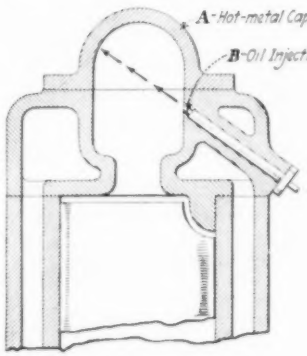


Fig. 11

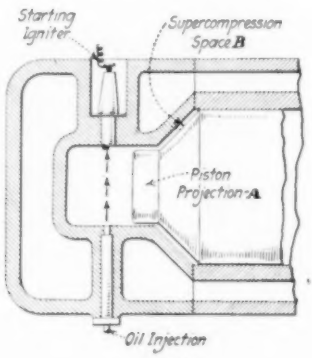


Fig. 12

Fig. 11 Hornsby Type of Combustion Chamber
Fig. 12 Crossley Cold Combustion Chamber

water-jacketed chamber connected to the cylinder by a neck, with an oil-injection nozzle *B* arranged so the jet strikes the hot cap to produce ignition by contact, is typical of a group of engines that started with the Hornsby.

COLD-WALL EXPLOSIVE-COMBUSTION SOLID-INJECTION ENGINES

37 As a result of a fairly general knowledge of the conditions surrounding these so-called solid-injection semi-Diesel engines, attention has been directed toward substitutes that would have some of the good qualities they had — simplicity, cheapness, foolproofness — as well as the good properties of the Diesel — cold walls, ignition by compression, and cleaner combustion with greater independence of fuel quality. Efforts to produce a cold-wall engine are directed along both lines. One is the explosive-combustion engine, the other,

the non-explosive-combustion engine. Cold-wall explosive-combustion solid-injection engines are comparatively new. The British Crossley (Fig. 12) has a piston with a conical end and a cylindrical projection *A*. The cylinder head is completely water-jacketed. As the piston approaches the head, the projection *A* will pass the corner of the cylinder head, at which time the air in the annular space *B* is trapped. This projection is a loose fit but not too loose, so that during the time it is passing into the head bore there is a violent annular stream of hot compressed air cylindrically distributed down the sides of the combustion chamber and back along the center. Into that stream of air is injected a fine spray of oil

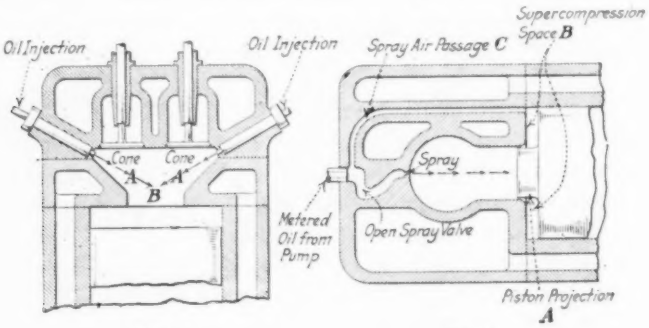


Fig. 13 Fig. 14

Fig. 13 Price-Ingersoll-Rand-De La Vergne Cold Combustion Chamber

Fig. 14 Haselwander Cold Combustion Chamber

that is instantaneously ignited, burning as fast as oil and air come together. The combustion is explosive in type but not in fact.

38 An American representative of the same class of engine is found in the Price construction (Fig. 13), which works differently and in which the combustion is normally explosive in fact. Here a fine spray is injected into a conical combustion chamber *A* on each side of a central cylindrical chamber *B*, up which a gentle air current rises during compression. This serves to help mix the fine spray with the air during the last of the compression stroke. Compression is adjusted so that the ignition temperature is reached just before the end, and as the air charge before compression is cooler at no load than at full load, the misfires that would happen, due to failure to reach ignition temperature, are prevented by an air throttle having the effect of retaining enough hot burnt products to avoid misfires. Should the air charge get too hot from any

cause, the whole charge might be ignited everywhere at the same time before compression was complete, producing a detonating combustion with explosive shock. Later injection would correct this if the charge were hot enough or compression high enough and make the action like that of the Crossley engine, where timing of injection is depended upon to prevent detonating shocks, as is spark timing in mixture engines burning gas or gasoline.

39 These engines are real modern improvements. They can, with a comparatively moderate compression — 200 to 250 lb. — give a fuel consumption that is substantially equal to the Diesel with its 450 lb. compression, but they must deal with a real difficulty. The combustion is essentially explosive combustion in fact or in type, and with explosive combustion not correctly timed — a little too early or too fast — detonations, either regularly or intermittently, are almost sure to occur. The shocks due to these detonations constitute one of the objections, and this has led other designers and investigators to devote their attention to a new class of solid-injection Diesel engines with non-explosive combustion in cold walls, the attractive features of which are less or no tendency to detonate, greater ease of maintaining correct combustion, and equally good fuel consumption.

SOLID-INJECTION DIESEL ENGINES WITH NON-EXPLOSIVE COMBUSTION IN COLD WALLS

40 An early attempt to eliminate the air compressor from Diesel engines was made by the German Haselwander, as shown in Fig. 14, which is practically a Diesel engine with air injection but without a compressor. An open type of air spray is combined with a piston construction embodying a cylindrical projection *A* that traps air in the space *B*. A passage *C* leads this air around to the spray nozzle. The end of the piston supercompresses part of the charge of air, driving it over the fuel and causing it to deliver a spray into the cylinder. The difficulty is that there is no control of timing. Any change in the leakage between the piston projection and the walls changes the timing, while carbon causes a binding action. There is also a tendency toward reverse flow on the outstroke.

41 A modification of this engine designed by Trinkler and built by the German firm of Körting is shown in Fig. 15. Trinkler added a small piston *A* in a cylinder connected with the main cylinder at both ends. Just at the right time the piston *A* was moved out by

a cam control so as to force air from the back end through the passage *B* to the spray nozzle. This little auxiliary piston *A* furnished the charge of supercompressed air for spraying the fuel, and it was timed like the old make-and-break ignition of the gas engine. The objection to it was that it tended to stick and stop the engine.

42 In Höflinger's proposed engine, Fig. 16, a small cuplike cylinder with a piston projects into the working cylinder. A passage for oil is connected below the piston and before the fuel is wanted it is deposited by the pump in that cup or bottom end of the small cylinder. Just when it is wanted a timing cam drives the piston down, compressing air on top of the fuel charge and expelling both through holes *A* in the side as an air spray. So far as is known,

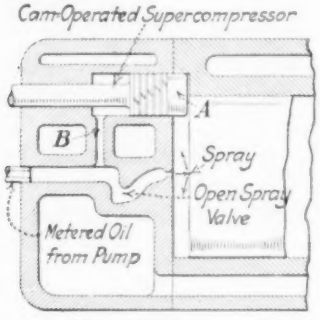


Fig. 15

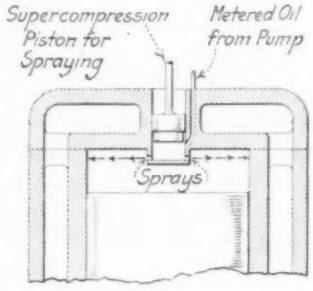


Fig. 16

Fig. 15 Trinkler-Körting Cold Combustion Chamber
Fig. 16 Hoflinger Cold Combustion Chamber

this engine has not been built, but it is very suggestive, especially when considered along with the very modern one reported from Detroit by Gernandt, Fig. 17, with special reference to use for automobiles. This has the Höflinger cup and fuel feed, but with a connection to a timing valve and to an auxiliary cylinder with a piston. The piston of Höflinger's cup is replaced by an independent piston *A* and separate timing valve *B*. Then at the right time the piston descends, the valve opens by a cam movement and the charge of air is supplied to the cup, spraying the oil into the cylinder.

43 Finally, the simplest of all in this class and the one form that has come into almost universal use for small stationary engines, and has practically no competition in its own field — farm units — is the Hvid, shown in Fig. 18. The Hvid retains the same fuel cup as the last two engines but with entirely different connections

for fuel and provisions for supplying and timing the spraying air. The fuel is delivered into the cup by gravity through a mechanical valve A, so that after it is in the cup the latter is a closed chamber. The fuel is delivered into the cup long before injection, in fact, before compression begins. During compression air flows into the cup through three holes, B, which prevent the outflow of oil prematurely, provided the holes are small enough so as to be capillary. After compression is complete, the air in the cup has a lower pressure than the air in the cylinder, which differential is quickly equalized on the expansion stroke. As soon as the cup pressure exceeds the cylinder pressure, the cup air will spray the oil into the cylinder. If the holes are too large, fuel will leak into the cylinder while the cup

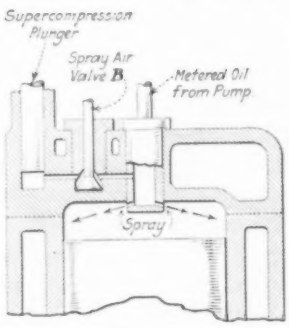


Fig. 17

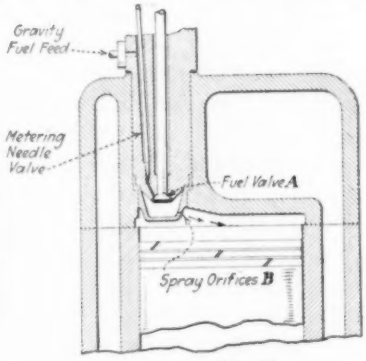


Fig. 18

Fig. 17 GERNANDT COLD COMBUSTION CHAMBER
Fig. 18 HVID-BRONS COLD COMBUSTION CHAMBER

is being charged. If they are too many, the air escaping into the cylinder will come out of some without spraying oil out of the others. There is a limit to both the number and size of the holes and the net result is that the Hvid cup seems to be limited to small cylinders, but in a small cylinder where a small number of capillary holes will pass the right amount of oil it is thus far supreme. It is the simplest possible, automatic in timing because cylinder pressures control the timing of the spray, and the only objection is that the combustion is a little late or slow. This little engine may be said to represent the limit of commercial success for small engines with solid injection supplied by air without an air compressor.

44 Steinbecker has devised another spraying scheme which has been worked out experimentally. In the form shown in Fig. 19 a very small bulb-like chamber with a narrow neck is connected to the

cylinder head. Through the top of this chamber a spray nozzle projects and delivers a jet or a coarse spray into the neck. At this time the air has already been compressed to and above the ignition temperature, so that as the oil escapes it immediately ignites, but not much can burn because not much of it can reach air in the narrow passage. What burns on the back face of the spray, according to Steinbecker, will raise the pressure in the bulb so as to produce an outflow of gases to respray finely what oil has been delivered to the neck and deposited on its walls. By the rise of pressure in the bulb, Steinbecker expected to really spray the main charge of oil.

45 Steinbecker has a recent engine, Fig. 20, in which the bulb,

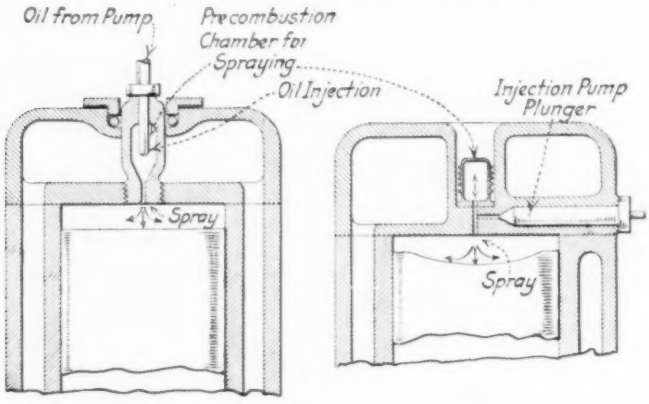


Fig. 19 Fig. 20

Fig. 19 First Steinbecker Cold Combustion Chamber
Fig. 20 Present Steinbecker Cold Combustion Chamber

chamber is retained, the neck passage is made extremely narrow so as to get a very high velocity through it, and injection pump delivery is led into the middle of the neck. It is timed so that oil enters near the end of the compression stroke while there is still some upward flow of air from the cylinder to carry it into the bulb chamber, which is unjacketed and hot, so as to produce ignition and combustion by explosion. The hot gases produced cause a reverse flow to the cylinder and spray the oil delivered later directly into the cylinder. This is Steinbecker's engine as it now stands, and as operated experimentally even in automobiles. The timing is directly by the pump, and should that pump force the oil in too soon, there would be an explosive shock. If the timing were wrong, the correct amount of oil would not enter the bulb and the spraying would fail also.

It is probably quite sensitive to pump timing, as is not the case with some others, including the Hvid.

THE DIVIDED COMBUSTION CHAMBER AS A MEANS OF PREVENTING DETONATING SHOCKS

46 One of the great difficulties with injection is to prevent the development of explosive shocks by too early, and loss of efficiency by too late, timing. Rapidly recurring detonations will wreck any machine in time. To direct the oil stream into the combustion chamber by a pump, without any other means of control of time and rate of combustion, is a method employed by Vickers in England, using a central pressure supply of oil admitted to spray valves by cam-timed oil valves. The equivalent was worked out by Junkers in Germany, using direct pump injection without timed oil valves, who succeeded in making it work in an aircraft engine, and to him is due the credit of first making a solid-injection heavy-oil engine that would fly in the air. In both cases, however — the German and the British — the fuel went directly into the cylinder and the production of detonating shocks was entirely a question of avoiding too early an injection — and earliness and lateness are a matter of a few degrees of crank angle. They must be extremely sensitive.

47 Here a different principle, intended to prevent detonating shocks and relieve the engine of the necessity for accurate timing, claims attention. In accordance with it the main combustion chamber is divided into two parts, one in the cylinder and the other removed — a divided combustion chamber. The air is compressed partly in the cylinder and partly in a connecting pocket, the major part in the former.

48 The first construction embodying this principle to be noted here is a Scandinavian one by Nydahl, shown in Fig. 21. He provided a spherical chamber connected to a cylinder by a series of holes and with an oil spray at the top. The piston stops at a point that leaves considerable air in the cylinder. There is a body of air in the cylinder and another body in the injection chamber at dead center. If the quantity of air in the injection chamber is small enough in comparison with that in the cylinder, then when the fuel is injected — the whole charge of fuel into the small amount of air — it cannot produce any explosive shock because so small an amount can burn. What, then, will happen to the rest of the fuel? It will change as it would in a gas producer where fuel reacts with less

air than is required for combustion. Some of it will burn; the rest will gasify; possibly some will merely vaporize. In this divided-combustion-chamber construction the piston becomes the principal element of combustion timing, somewhat as in the Hvid engine because the main combustion is produced by the flow from injection chamber to cylinder. The divided-combustion chamber is the principal element in preventing the explosive shocks, the piston movement controls the completion of combustion, and if the compression is high enough the whole structure can be jacketed.

49 Such a water-jacketed injection chamber, but of more or less cylindrical form, and embodying the divided-combustion chamber and side injection of fuel, is shown in Fig. 22. This is a Danish

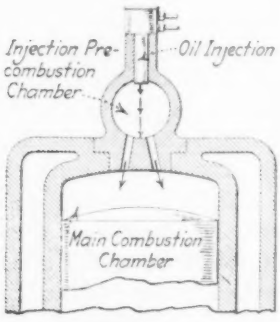


Fig. 21

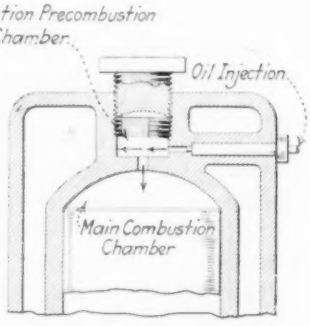


Fig. 22

Fig. 21 NYDAHL DIVIDED COLD COMBUSTION CHAMBER
Fig. 22 NIELSEN DIVIDED COLD COMBUSTION CHAMBER

form, by Nielsen, who also provided for adjustment of injection-chamber volume by a special cover or cap.

50 Still another type brought out in the course of development of the divided combustion chamber is the Leissner, Fig. 23. Leissner adds to the Nydahl or Nielsen injection chamber a tube which has holes in the sides and bottom to form the old-fashioned fuel distributor used with air- and solid-injection spray valves. The Leissner tube comes up close to the injection spray nozzle which delivers the oil inside the tube. Just as the small injection chamber alone prevents development of explosive shocks by limited contact of air and fuel, so does insertion of this tube add something to further limit the contact control. Leissner specifies that the holes through the bottom and sides of the tube, and the space above it, shall be so related to each other in area as to produce the following

series of actions: Compression carried first to ignition temperature; injection inside the tube in the injection chamber and partial combustion in the tube, producing a rise of pressure in the tube, which in turn projects jets of still unburned oil sidewise into the air around the tube. Combustion of these jets raises the pressure outside of the tube and causes reversal of flow back into the tube and down through it to the cylinder, helped by the movement of the piston, the air left around the tube finally passing through the tube and expelling the fuel charge in front of it into the cylinder, the space around the tube and in the tube being in series. This is a new engine brought out in Sweden, and it is now being introduced into this country.

51 Finally, in this class of divided-combustion-chamber in-

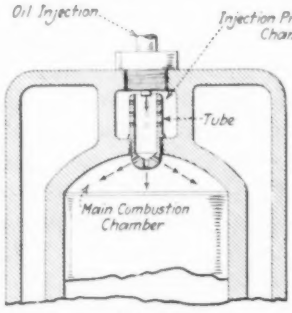


Fig. 23

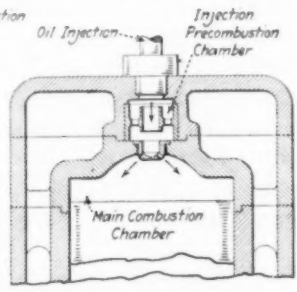


Fig. 24

Fig. 23 LEISSNER DIVIDED COLD COMBUSTION CHAMBER
Fig. 24 WORTHINGTON DIVIDED COLD COMBUSTION CHAMBER

jection engines there is a new one developed by Worthington — the class in which the injection or precombustion chamber is used to limit the development of explosive shocks and connected to the cylinder by an ejection orifice through which gasified fuel is expelled to the cylinder by a pressure differential due partly to the movement of the piston and partly to the precombustion. This is shown in Fig. 24. It has a tube open at both ends and supported from the side walls with wide spaces at each end. The bottom of the injection chamber has a passage to the cylinder of the usual fuel-distributing sort, forming a fuel-ejection orifice. There are large holes through the tube-supporting web, so that the injection chamber acts as a single chamber, with the tube acting as a sort of fuel baffle or guard. The charge being injected into the tube is limited by the tube as to air contact. The tube constitutes a fuel guard, by means of which the amount of precombustion can be controlled and at

the same time the fuel is gasified in the hot limited amount of air. A rise of pressure in the injection chamber, due to partial combustion, is supplemented by depression of the pressure on the cylinder side due to the piston movement, which starts the flow of parallel streams of air and fuel toward the cylinder. This action creates in effect a bunsen burner in the top of the cylinder. This arrangement gives a construction practically as simple and foolproof as the little Hvid, except for the injection pump, but one that seems to be adapted to a wider range of cylinder sizes and speeds and less limited by fuel quality.

52 Much experimental research has already been completed and more is planned, the results of which will be presented in a later paper.

No. 1826

REQUIREMENTS IN THE DESIGN OF STEAM POWER STATIONS FOR HYDRAULIC RELAY

By E. B. Powell,[1] Boston, Mass.
Member of the Society

SUPPLEMENTAL power in some form is an essential to the commercial development of our remaining unused eastern water powers and, with the possible exception of the Niagara-St. Lawrence system, an essential to the further development of such of these water powers as are now utilized in part. The nature and extent of this supplemental, or relay, power will be governed in a large measure by the character of the stream and the amount, character and location of the load and the importance of continuity in power supply.

2 In many instances the required relay, at least sufficient for the initial development, may advantageously be provided in the stream itself by adding reservoirs, or merely pondage, to give some artificial control of the flow. In other instances the electrical interconnection of water-power systems, so pooling the water resources of dissimilar streams, may also afford sufficient initial relay in hydraulic power. In general, however, the fullest commercial utilization of water power can only be had by the aid of independent relay, which in the greater part of the country today is most satisfactorily and economically provided in the steam power station.

3 The capacity and design of such a relay station should be decided as closely as may be by the particular functions it is intended to perform. The functions of the relay, while they may vary widely, are of two general types, flow or head deficiency make-up and emergency reserve, the second being supplemental to the first. For

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either type of service the capacity required in any particular case will depend upon the amount and characteristics, initial and prospective, both of hydraulic development and of load. The design, while broadly determined by the usual factors, such as location, power market, water and fuel supplies, fuel costs, which control in the case of the independent central station, is in general, on account of the commonly low load factor, less dependent upon considerations of operating economy. On the other hand, character of load and dependability of delivered hydraulic power are factors of prime importance and, for proper realization of the economic possibilities of the development as a whole, the design should also be governed in many important features by the characteristics of the hydraulic development and the relation of that development to the load.

THE FLOW OR HEAD DEFICIENCY TYPE OF RELAY

4 Considering first the station intended solely to make up the power deficiencies of the hydraulic development, its functions may include any one or more of the following: seasonal operation to make up deficiencies in hydraulic power from low water or flood; absorption of load growth between stages of hydraulic development; operation as the main source of power. As reflected in the functions of the relay station and as affecting its design, hydraulic developments may be grouped as of four general classes:

I. Minimum Flow Development. Developed hydraulic capacity equal to or exceeding maximum system load. Resultant minimum available flow only slightly, and for brief period in the year, below corresponding load requirements. Reserve provided in hydraulic station.

II. Medium Flow Development. Developed hydraulic capacity equal to or exceeding maximum system load. Resultant minimum available flow below load requirements for considerable period, 40 per cent or more, of the year. Reserve provided in hydraulic station.

III. Continual Relay Development. Available hydraulic capacity slightly below system load requirements throughout greater part of the year.

IV. Supplemental Development. Hydraulic development merely an adjunct to the steam power station, which carries the bulk of the load.

The type of reserve referred to in Classes I and II is for relay against failure of equipment within the hydraulic station itself. Under the conditions outlined, this reserve may often advantageously be installed as part of the hydraulic station rather than in the steam station. As is well known, from 50 to 75 per cent of the cost of the average hydraulic development is fixed and independent of the capacity installed, so that sufficient capacity for this type of reserve may frequently be included at comparatively low unit cost.

5 *Effect of Growth of Load.* It will usually be found that the conditions represented by Classes I, II, and III are merely stages in approach to those of Class IV. This is especially true in the East. However, the change in relative status of the hydraulic and steam power stations may be very slow, particularly so where a series of hydraulic developments may be brought in successively, as warranted by load conditions; in which case to pass beyond the conditions of Class II may require a matter of decades. For these reasons the entire project should be studied broadly in the beginning both as to the ultimate physical possibilities and limitations of the hydraulic development, or developments, and as to the character and probable growth of the market for power. Neglect of such basic analysis in the initial planning of the development risks serious financial loss either in investment or in operating costs.

6 *Seasonal Variation of Flow.* The seasonal flow of the Hudson River in New York is so variable that the hydrographs of two successive years, 1915 and 1916, with their corresponding deficiency curves may be used in illustrating developments of all four of the classes just referred to. In Fig. 1 it will be seen that if this hydrograph is assumed to represent the year of minimum flow for a particular stream and if a flow of about 3000 sec-ft. would supply the maximum power requirement, a development providing proper equipment reserve at this flow would approximate the conditions of Class I. If growth of load raises the daily power demand above the capacity of the available minimum stream flow and this growth of load is accompanied by a corresponding increase of generating capacity of the hydraulic station to utilize, say, 8000 sec-ft. with proper reserve, the conditions become those of Class II. Turning to Fig. 2, development of the hydraulic plant to utilize a flow of, say, 10,000 sec-ft. to 12,000 sec-ft. with daily load demands somewhat in excess of the mean available flow, would give the conditions of relay service under Class III. Continued growth of load with

a single hydraulic development as applied to either type of stream flow will ultimately bring about the conditions of Class IV.

7 *Reservoir Storage and Pondage.* Of course, in the case of

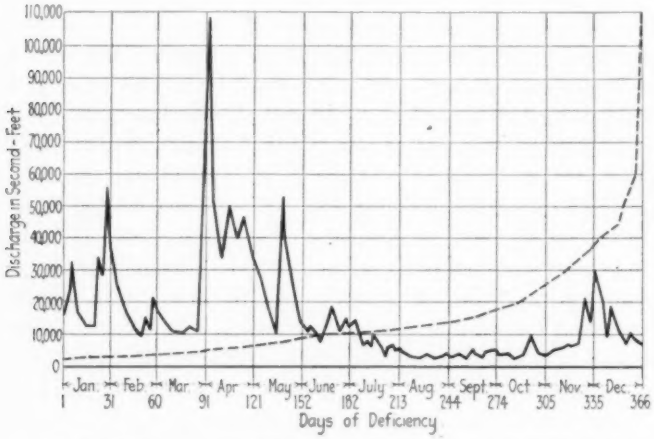


FIG. 1 DAILY HYDROGRAPH AND DEFICIENCY CURVE FOR 1916, HUDSON RIVER AT TROY DAM, NEW YORK

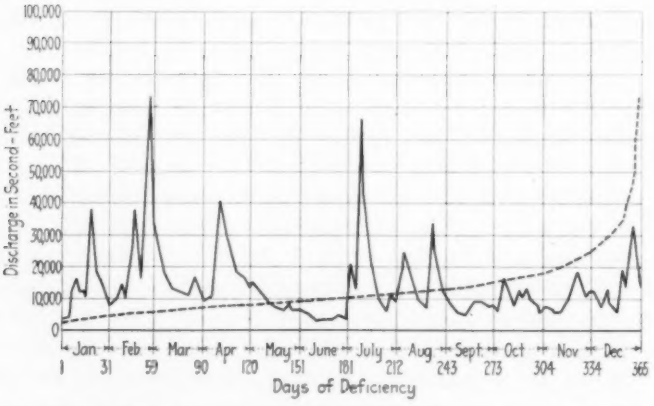


FIG. 2 DAILY HYDROGRAPH AND DEFICIENCY CURVE FOR 1915, HUDSON RIVER TROY DAM, NEW YORK

many streams, it would be feasible to provide sufficient reservoir storage to smooth out the stream flow to a very material extent, thus raising the minimum or primary capacity of the potential development beyond the maximum load demands considered above under

Classes I and II at least, and possibly beyond that considered under Class III. For purposes of the present discussion, however, reservoir storage, as distinct from pondage in the usual sense, will not be further considered.

8 The effect of pondage upon the relay station requirements may be judged from reference to Fig. 3, which shows a typical manufacturing-town load with the "per cent load" curve superimposed for convenience of interpretation. If the minimum flow is assumed equivalent to 50 per cent of the day's energy requirement,

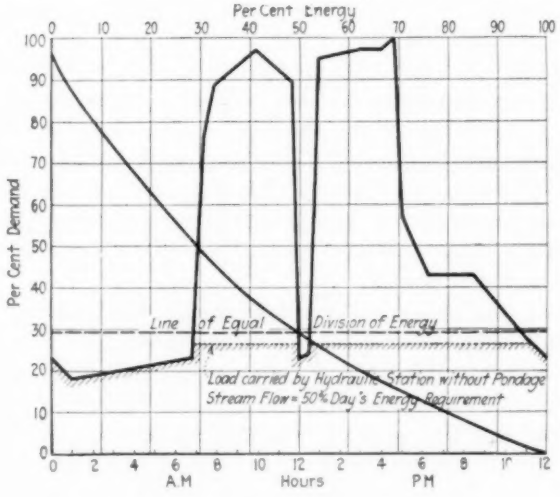


FIG. 3 TYPICAL MANUFACTURING-TOWN LOAD

(24-hour load factor, 53 per cent.)

with full 24-hour pondage provided, the hydraulic station could take off the upper 70 per cent of the demand, permitting the relay station to carry the base load at a daily load factor approximating 90 per cent with a demand only 30 per cent of the total. On the other hand, in the absence of pondage the relay station must be designed to carry about 75 per cent of the total demand, and on this basis its daily load factor would be reduced to about 40 per cent. It is apparent that, in the absence of pondage, higher capacity will in general be required in the relay station, and the relay-station load factors, both daily and annual, will tend to lower values than where full pondage is provided.

RELAY STATION FOR MINIMUM FLOW DEVELOPMENT

9 Referring again to Fig. 1, it will be seen from the deficiency curve that if the maximum load demand is within the capacity of the 3000-sec-ft. flow which is the developed capacity considered for Class I, the relay station except for conditions of equipment failure should not be called upon for operation in excess of 5 per cent of the year. It is not uncommon for relay stations under such conditions to stand idle an entire year. Obviously, for service of this character the actual operating economy of the relay station is of relatively small moment. The important considerations are low fixed costs and dependability.

10 As the load demands increase and the operating conditions for the relay station approach those of Class II, steam-power production costs are of slightly greater moment but in general are still tremendously outweighed by fixed charges and other costs independent of production. Usually it is only when the conditions of Class III are reached that the annual load factor of the relay station has risen to a value sufficient to make fuel economy as such an important factor in design.

11 For the extremely low load-factor conditions of Class I type of equipment, dimensions and arrangement must all be studied with a view to attaining the desired output with minimum investment compatible with simplicity of station and low cost of attendance. Type of combustion equipment and the arrangement of furnace must of course be governed by the character of fuel available.

12 Coal in its more ordinary form will in general be the most satisfactory fuel. The stoking equipment should be of the forced-draft type for high capacity, either underfeed or chain grate, depending upon the class of coal. Draft facilities should be designed to take care of the maximum rate of combustion that can safely be maintained by furnace and grate over the period of peak load on the station. Economizers cannot be justified for the conditions of Class I, but it may be advisable to make provisions for their later installation to meet higher-load-factor conditions in the future.

13 The selection of type and size of boiler, decision as to total rated boiler capacity to be installed, and the proportioning of combustion equipment to boiler heating surface must take into account not merely the cost of that equipment only, but also the costs of building and all related equipment such as fuel- and ash-handling facilities, and mechanical draft. The cost curve of Fig. 4 shows

that, for the particular case considered, when all related factors are taken into account boiler heating surface may be installed to the comparatively high ratio to grate area of 50:1 without appreciable increase of total boiler-plant cost. Higher ratios are accompanied by some increase of cost. The use of a lower ratio will result in needless waste of fuel.

14 Equally broad considerations should be applied to the design and proportioning of the condensing equipment. Considering

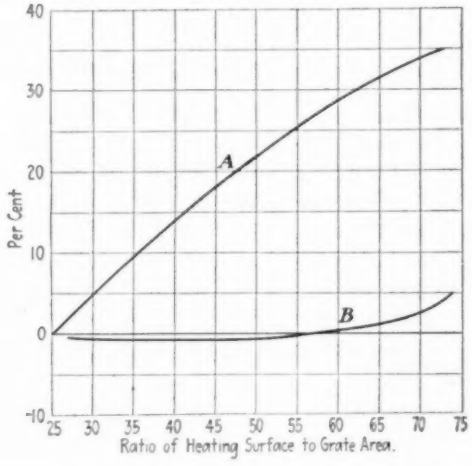


FIG. 4 BOILER-PLANT CONSTRUCTION COST AND CAPACITY AS AFFECTED BY RATIO OF BOILER HEATING SURFACE TO GRATE AREA

A — Increase of steam capacity per sq. ft. of grate area
B — Difference in cost of boiler plant per unit of steam capacity
(Costs and capacities expressed as percentages based on ratio of boiler heating surface to grate area of 25:1 for a particular case.)

the net increment cost of the boiler plant per unit of effective steam capacity and the increment cost of condensing plant for unit change of vacuum at the turbine exhaust, the condensing equipment should be designed to give the degree of vacuum under conditions of the required maximum load which will result in the lowest construction cost for the plant as a whole. In Fig. 5, which deals with a particular set of conditions, curve *A* shows the relative construction costs for condensing equipment to attain different degrees of vacuum, the cost of the 29-in. design being taken as 100 per cent; curve *B*, drawn to the same scale, the increase in boiler-plant cost resulting from the increased total steam demand which is had from different

degrees of vacuum less than 29 in.; and curve *C*, which is a summation of *A* and *B*, still expressed in terms of 29-in. condensing-plant cost, the resultant effect upon total station cost, of designing the plant for different degrees of vacuum.

15 These two charts, Figs. 4 and 5, are correct, as to detail, for the particular conditions only for which they were estimated, but they clearly illustrate the interdependence of equipment costs

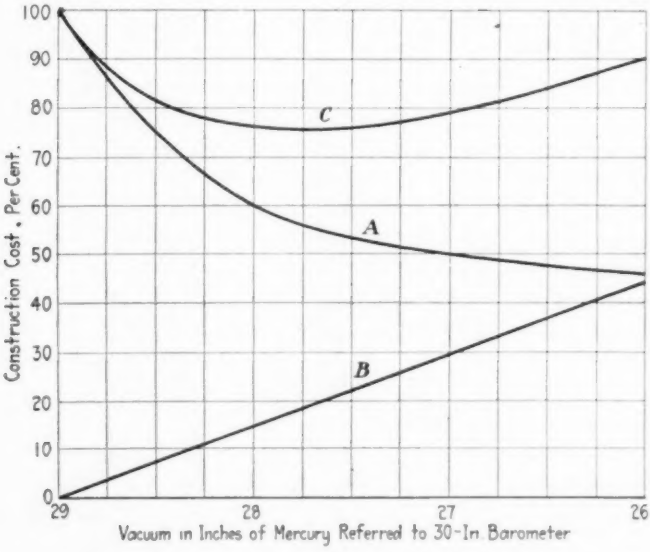


FIG. 5 RELAY STATION CONSTRUCTION COSTS AS AFFECTED BY CAPACITY OF CONDENSING EQUIPMENT

A — Cost of condensing equipment
B — Increase in cost of boiler plant
C — Resultant effect upon total station cost
(All costs expressed as percentages based on cost of condensing equipment designed for 29 in. vacuum and 70 deg. cooling water for a particular case.)

and the importance of taking all related factors into account in deciding upon the type and size of apparatus. Similar considerations should be applied to the selection of generating units and to the determination of steam pressure and temperature.

16 Instrument equipment for mechanical apparatus such as boilers and condensers should be complete in so far as required as guides to efficient combustion and to the attainment of the highest degree of vacuum of which the condensing apparatus is capable; not that efficiency in the sense of low fuel rate is of special importance

in this service, but that it is an essential to the attainment of maximum output from the major equipment installed which is the prime purpose of the station design.

17 It will usually be found advantageous to provide electric drive for one complete set of essential auxiliaries to permit starting these without waiting for steam. The main electrical features of the relay station for the Class I conditions will be governed largely by the relation of the station to the rest of the system, provision being made to compensate for line or load characteristics wherever required.

RELAY STATION FOR HIGHER-LOAD-FACTOR CONDITIONS

18 As the system load grows, if the period of relay station operation is permanently lengthened, taking the development into the conditions of Class II for example, the higher load factor on which it may be possible to operate any new equipment may justify greater consideration of actual operating economy, condensing equipment may be of more liberal design, and boiler capacity may be added in greater ratio to steam demand, so permitting reduction of combustion rates.

19 Further growth of load and extension of the operating period for the relay station into the conditions of Class III will justify in new equipment still further consideration of operating economy; until, as the conditions of Class IV are reached, practically the same factors govern as in the usual independent type of central station.

20 As may be inferred from this outline of the progressive extension of the relay station to keep pace with the system's requirements, the relay station properly designed for low-load-factor conditions may be readily converted to a high-load-factor station, and so converted should operate as such with but a fraction higher total costs than the station specially designed for the higher load factor.

21 Fig. 6 shows the power costs from two types of station operating on the same conditions of load. Station A is designed initially for relay service under Class I conditions. Improvement in load factor is accompanied by the installation of such additional equipment and of such economic characteristics as may be warranted. Station B is designed for continuous operation at 50 per cent annual load factor. Its first cost is nearly 50 per cent higher than that of the initial step of station A and, while more economica at the load factor for which it is designed, on the lower range of load

factors the higher efficiency of its equipment is insufficient to balance the higher fixed charge. Accordingly, where there is doubt as to the actual load factor at which the relay station may be called upon to operate, it will in general be advisable to design for the minimum probable load factor rather than risk unnecessary expenditure in providing for load conditions that may require years to attain.

22 If the fundamental design is right, the initially high operat-

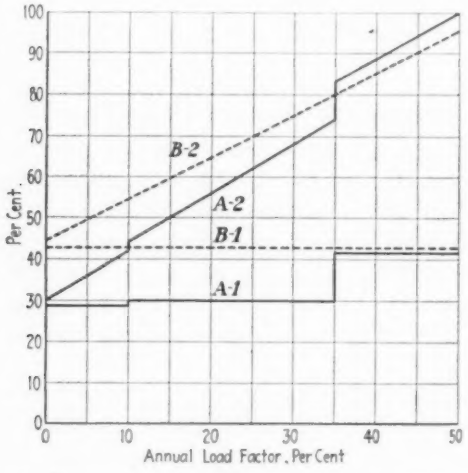


FIG. 6 ANNUAL POWER COSTS OF STEAM RELAY STATION AS AFFECTED BY LOAD FACTOR FOR WHICH PLANT IS DESIGNED

A-1 — Annual fixed charges
A-2 — Annual total power cost } Class I Relay Station
B-1 — Annual fixed charges
B-2 — Annual total power cost } 50% Load Factor Station

(All costs expressed as percentages based on total power cost for Class I relay station developed for 50 per cent load factor and operated at 50 per cent load factor.)

ing charges characteristic of the low-load-factor station are readily outgrown, whereas the fixed charges, essentially higher for the high-load-factor station, are inescapable and remain until amortized a permanent burden on the development.

EMERGENCY RESERVE TYPE OF RELAY

23 Turning now to the station which must also function as an emergency, or breakdown, reserve, the emergency service required of such a station may vary from floating capacity for instant availability to merely providing against interruptions of several

hours or days. The requirements in any particular case will be governed in a large measure by the character and importance of the load served.

24 Miscellaneous power service and lighting should, in general, be completely relayed against more than momentary interruption; for some classes of power, frequently that applied to electrolytic processes, for example, longer interruption of service is not of so serious moment, not sufficiently serious to warrant the expense of full relay; on the other hand, an extensive d.c. network, such as that in large metropolitan cities, must be absolutely safeguarded against interruption, however brief.

25 When the steam relay station is to serve as an instantaneously available reserve, the necessary excess capacity in generating units must be maintained on the line and actually under steam at all times. Station design to provide most economically for such service must afford, for minimum investment, a maximum of flexibility in capacity with minimum stand-by charges. Characteristics in equipment peculiarly favoring these requirements are:

1 Combination electric-steam drive on the larger auxiliaries, such as circulating pumps and mechanical-draft fans, for the units operated as floating stand-by
2 Turbo-generating units with direct-driven exciter, and with overspeed-controlled steam bypass for turbine-blade cooling independent of normal governing valves
3 High ratio of turbo-generator capacity in comparison with point of lowest water rate
4 Wide flexibility with quick response in control of boiler output and rate of combustion
5 High ratio in capacity of combustion equipment vs. area of boiler heating surface.

26 Where commercially available, fuel oil with mechanical atomization is particularly advantageous on this class of service, and its use permits the fulfilment of these boiler-plant requirements to an almost ideal degree. With proper design the response to control is practically instantaneous; capacity attainable is the highest of any commercial fuel. Oil fuel has the added advantage, particularly important in stand-by service, of minimum banking loss.

27 Pulverized coal, bituminous or sub-bituminous, for the larger plants, is only second to oil in flexibility of control and low banking loss. In general the underfeed type of mechanical stoker, given

proper ratio of grate area to heating surface, will also fully meet the requirements in both flexibility and capacity, but banking losses are necessarily higher than for either oil or pulverized coal.

28 The class of fuel to be adopted in any particular case should be governed by the local conditions of fuel market, load and relay-station load factor, present and prospective, station location and capacity. However, it is usually true that fitness of equipment, and of its arrangement, for the requirements of the fuel determined upon, are more important than selection of type of fuel.

29 Emergency relay service on loads for which absolute continuity is of less vital importance will normally involve actual readiness to serve only during periods of anticipated power deficiency such as between seasons when stream flow is precariously near the actual load requirements and any slight diminution of flow will necessitate calling upon the relay station, or when there is possibility of sudden ice accumulation, or threatening floods.

30 For relay service of this character it will usually be sufficient to maintain a few boilers under fire and the larger of the essential auxiliaries turning over. The general characteristics of equipment outlined for conditions requiring instantaneous availability of the reserve apparatus will be advantageous in this case also. The combination electric-steam drive for auxiliaries may be omitted in the interest of first cost as the relay is not necessarily intended to meet full failure of hydraulic power, but the provision of electric drives for one complete set of essential auxiliaries, as recommended for the normal deficiency make-up of relay station to permit prompt starting of such equipment without waiting for steam, will usually be found particularly advantageous under the service conditions now considered. It may also under some conditions be advantageous to make use of electrical heaters in maintaining hot-water circulation in certain of the boilers, instead of actually holding them under fire. Oil is again the ideal fuel, but it is questionable if in the average case pulverized fuel will possess any advantage over the forced-draft underfeed stoker. New developments may, of course, readily change the relative status of these two methods of burning coal. In general the features of design which are specially advantageous for this type of relay are common to the instantaneous type of relay and are readily embodied in the station designed for the simpler type of make-up service.

31 Considering again the general characteristics and requirements of the steam relay station, the feature which should be re-

garded as secondary only to low construction cost and adequate dependability is that of low attendance requirements, and not merely for the non-operating period but for actual operation as well. With this object in view the design should tend to large units and, as far as compatible with thorough simplicity, to automatic control. Where feasible, combining the functions of distributing station with those of the hydraulic relay assists materially in holding down both construction costs and attendance for the relay station. It should be borne in mind, however, that maximum returns on the investment in any type of steam power station can only be had through skilled and well-trained operators. Such men cannot be picked up on a moment's notice. It is true that it is often possible during periods of plentiful water supply to distribute men from the steam stations through other departments, but the number of men that may actually be employed in this way is limited. Where men are held over long periods without really effective employment, there is invariably loss of efficiency and of morale.

No. 1827

APPLICATION OF THE LAW OF KINEMATIC SIMILITUDE TO THE SURGE-CHAMBER PROBLEM

By W. F. Durand, Stanford University, Cal.
Member of the Society

The problem of the hydraulic surge chamber gives rise to certain differential equations which do not admit of solution in analytical form. Under these conditions, various modes of treatment have been proposed. The present paper develops and describes a comprehensive experimental method utilizing the law of kinematic similitude. By this method a laboratory model of convenient dimensions is set up and subjected to a program of operation corresponding to that for the field installation. The movements of water level, times, etc., as measured for the model, are then multiplied by suitable relation factors, thus giving the results to be anticipated for the field installation.

The model may thus, in a sense, be considered as a form of mechanism which solves the equations as applied to the model form, and these results multiplied by suitable relation factors give the corresponding results to be anticipated in the field case.

The principal advantages of the model method are as follows:

1 The model is simple and readily set up. Once in operation, the most varied conditions may be quickly examined, thus making possible a range of examination in detail, the extent of which would be quite impracticable by any methods of numerical computation.

2 In all analytical methods of treatment, pipe-line friction has been assumed to vary with the square of the velocity. In the model method the friction may be taken to vary with v^n, where n is usually found to be about 1.85.

3 In analytical methods the surge chamber is usually taken as uniform in cross-section. With the model, the chamber may be of any form whatever.

4 The model method is readily extended to the case of a surge chamber with spillway or to the case of multiple chambers on one pipe line.

5 The model method provides readily for the examination of periodic fluctuations in power demand, of any amplitude and any periodicity. The examination of such fluctuating programs is beyond practical reach by numerical methods.

IT IS assumed in what follows that the general physical phenomena of surges and of surge-chamber operation are familiar to the reader. They have been often described both in textbooks on hy-

Presented at the 1921 Spring Meeting of the National Academy of Sciences Washington, D. C.

draulics and in special papers on the subject, and to such sources the interested reader may refer for more complete details.[1]

2 The problem of the surge chamber as a problem on hydraulics gives rise to a pair of simultaneous differential equations as follows:

$$\frac{L}{g}\frac{dv}{dt} = H - y - cv^n \quad \ldots\ldots\ldots [1]$$

$$-\frac{F}{A}\frac{dy}{dt} = v_2 - v \quad \ldots\ldots\ldots [2]$$

where L = length of conduit
v = velocity
H = head measured from datum at bottom of chamber to level of water in supply reservoir
y = height of water level in surge chamber above datum at bottom of chamber
F = cross-section area of surge chamber
A = cross-section area of conduit
c = factor such that cv^n = sum of friction and velocity heads at velocity v
n = index in expression cv^n and usually found numerically to be about 1.85.

3 Regarding the factor c in Equation [1] it may be noted that the sum of the velocity and friction heads would usually be represented as the sum of two terms of the form $v^2/2g + bv^m$, where m is the index of variation for frictional resistance with velocity and b is the factor connecting v^m with loss of head due to friction and depending, primarily, on length of line, mean hydraulic radius, and character of surface. It is, however, simpler to combine these two terms into one (the first is in all practical cases relatively small), and it is quite as easy to represent the sum, as derived experimentally, by a single term of the form cv^n as it is for the friction head alone by a term of the form bv^m. In any actual case, furthermore, the values of m and n would be hardly distinguishable. The value of n thus indicated is usually found to be about 1.83 or 1.85. For very rough surfaces it will approach more nearly to 2.

4 It should be further noted that the actual details of the transition phenomena during the period of change of load and while settling down to new conditions will depend on the character of the

[1] Trans. Am. Soc. M. E., vol. 34 (1912), p. 319; *Western Engineering*, December 1913.

governor action during the same period. At least four assumptions may be made regarding the results of such action, as follows:

(*a*) A uniform rate of flow at the amount required for the new power demand under steady flow conditions
(*b*) Constant power output from the wheel
(*c*) Constant power input to the wheel
(*d*) Control valve or gates put immediately into the position corresponding to the final steady flow condition and left unchanged during the transition period.

5 No one of these assumptions is accurate. If we could assume the load demand to instantaneously change and then remain uniform at the new value and if we could assume the governor capable of exercising perfect and complete control, then it would, under the new conditions, maintain constant speed and hence constant power.

6 No governor is, however, able to operate in this ideal manner, and the load changes are not usually instantaneous followed by uniformity at the new value. Actually the load may change suddenly, followed by minor variations in settling to something approaching a uniform value. The governor is, furthermore, chasing back and forth with a time lag between speed changes and the actual operation of gate or valve mechanism. Any attempt to include governor action in the equation for the movement of the water in the surge chamber can thus be only partial at the best, and such attempt greatly complicates the resulting equations and their treatment, no matter by what method.

7 Equations [**1**] and [**2**] are based on assumption (*a*), that of a constant-volume flow at the rate which, under steady conditions, would serve to give the power required under the assigned new conditions of power demand, and this assumption results in such a notable simplification in the analytical expressions involved that mathematical discussion of the problem is usually based on the equations in this form.

8 It does not follow, however, that for the application of the law of similitude this particular assumption is the one best suited to the experimental phase of the problem. As will be shown at a later point, assumption (*d*) is by far the most convenient for experimental realization. However, the determination of the conditions for similitude and of the numerical ratios through which the various relations are expressed is quite independent of the particular assumption which may be made regarding the results of governor action

on the flow conditions during the period of change. Hence, since assumption (*a*) leads to the simplest system of Equations [**1**] and [**2**], we may conveniently use these equations for the further development of the general problem, and then at a suitable point return to the question of the consequences to be anticipated from these various assumptions regarding the results of governor action during the period of change.

9 As treated in the literature of the subject, furthermore, the index n is usually taken as 2. This corresponds in effect to the assumption that friction loss varies as the square of the velocity — an assumption which is well known to be definitely in error. The difficulties of analytical treatment in any form are, however, so

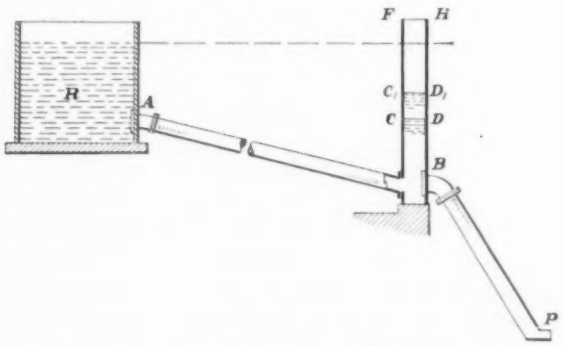


FIG. 1 DIAGRAMMATIC REPRESENTATION OF RESERVOIR, CONDUIT AND SURGE CHAMBER

vastly increased by the assumption of a value of n such as 1.85, that the value 2 has been commonly accepted for the index of v.

10 The two equations with the value $n = 2$, when combined, give rise to a differential equation of the second order and second degree and containing both the first and second powers of dy/dt. This equation does not seem to admit of analytical solution, at least in terms of present known functions.

11 Some years ago the author made brief mention of the possibility of treatment by experimental methods making use of the principles of kinematic similitude,[1] and at a subsequent time [2] a discussion of the problem was published covering the application of this method to the case with the index $n = 2$.

[1] Trans. Am.Soc.M.E., vol. 34 (1912), p. 319.

[2] *Western Engineering*, December 1913.

12 The present paper aims to generalize this mode of treatment by extension to the case with a general index n, and including likewise the possible case of two or more surge chambers connected to the same line.

13 It will be evident that if the equations with $n = 2$ do not admit of analytical solution, any such mode of treatment with $n = 1.85$ will be likewise and still more definitely out of the question, and if any general mode of treatment is to be found, it will needs fall under one or the other of two classes: (1) numerical, by the method of numerical quadratures, so-called; or (2) experimental, by the use of a model connected with the actual field case through the principles of kinematic similitude. It is with the latter of these that the present paper deals.

KINEMATIC SIMILITUDE

14 Before attempting to develop the conditions of kinematic similitude between two cases, it is necessary first to note what is implied by this term.

15 To develop the idea, let Fig. 1 denote diagrammatically the characteristic elements of a given case. R denotes the reservoir supplying water through the conduit AB to a power plant at P. Let BFH denote a standpipe or surge chamber relatively near to P. Let the plant be running at a steady rate of output with a velocity of flow v_1 in AB. Then, taking the case of demanded load, let there come a sudden demand for more power and consequently more water such as would require, under steady conditions in AB, a velocity v_2. The conditions of flow in AB will, however, not permit of an immediate response. The proximity of the surge chamber to the power house does, on the other hand, permit of prompt or sensibly immediate response and accordingly the additional amount of water required to make up the supply to the wheel, corresponding to a velocity v_2 in AB, is drawn from BFH. In consequence the level in BFH falls, occupying at a given instant of time immediately after the new power demand, some level such as CD, while the level for steady-flow conditions still remains sensibly at C_1D_1. This difference of level, the actual level lying below that for steady conditions, means the development of an accelerating head measured by the difference between these two levels. In response to this accelerating head, the velocity of flow in AB will begin to increase and such increase will continue so long as the accelerating head remains operative — that is, so long as the actual level lies below that for steady-

motion conditions. In consequence the velocity v will pass gradually through a series of values on the way from v_1, the initial value, to v_2, the final value. In general, during this period the actual level will lie below the level for steady conditions with the actual velocity v, thus constituting a continuing accelerating head and in answer to which the water in AB undergoes a corresponding positive acceleration. If the conditions should be such as to bring about a gradual decrease of this accelerating head as the velocity approaches the value v_2, ultimately vanishing when $v = v_2$, then the series of transition phenomena comes to an end with this contlition, and the water level remains at its new location with velocity v_2. Actually, however, these conditions are not fulfilled. When the velocity first reaches v_2 the acceleration head is still positive; that is, the actual level is below that for steady conditions with velocity v_2, and in consequence the velocity is carried on to some value greater than v_2 until finally the two levels do come together, but with a velocity $v > v_2$. This means more water is coming along AB than is required at the power plant, and in consequence the excess will be received by the surge chamber and the actual level of the water will rise above its location for steady-flow conditions, thus producing a *retarding* head and gradually reducing the velocity v from its maximum value back toward v_2 again. On reaching v_2, however, the actual level will still lie above that for steady-flow conditions, a corresponding retarding head will still be operative and the velocity will be carried beyond and below v_2 until the retarding head vanishes and becomes replaced by an accelerating head, when the velocity is again carried upward toward the value v_2. In this way there will be an indefinite number of oscillations of level about the final location and a corresponding indefinite series of oscillations of value of the accelerating head from positive to negative and back again, and a corresponding series of fluctuations of velocity above and below v_2 — all of these oscillations and fluctuations of continuously diminishing amplitude, until finally they all sensibly die out together with the velocity at v_2 and the water level at the corresponding location for steady conditions.

16 In the case of rejected load and a sudden excess of water flowing in AB, with v_2 less than v_1 the excess water will be received by the surge chamber and will result in a rise in the actual level above that for steady-flow conditions, thus producing a retarding head. In answer to this, the velocity will become gradually *reduced* from v_1 toward v_2, passing through a series of fluctuations, with

corresponding oscillations in the water level, in manner entirely similar to that for demanded load.

17 It will then be evident that if we should represent graphically on a time axis the essential phenomena which characterize this transition period, we should have curves as follows:

1 A time history of the movement of the actual water level CD

2 A time history of the movement of the level for steady conditions with the actual velocity v

3 A time history of the difference of these two levels, constituting in general (for demanded load) an accelerating head, and measured by the left-hand member of Equation [1]

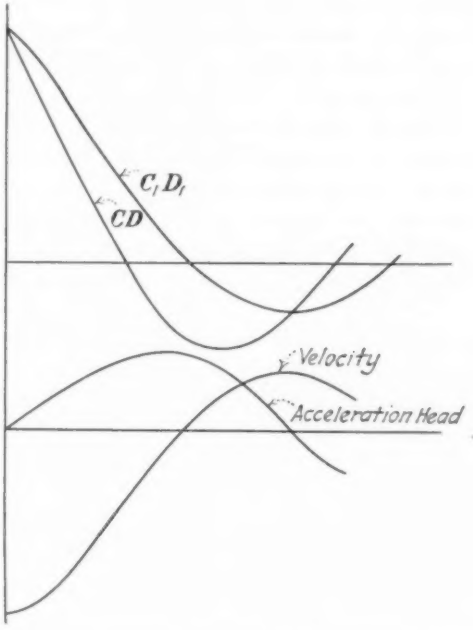


FIG. 2 CURVES REPRESENTING ESSENTIAL PHENOMENA CHARACTERIZING FIRST PART OF TRANSITION PERIOD

4 A time history of the velocity v as it passes through periodic fluctuations from the initial value v_1 to the ultimate value v_2.

18 Suppose such a group of curves for the first part of the transition period be as represented in Fig. 2. Let us designate the physical system to which these curves correspond as system A. Now let us imagine a system B, comprising the same elements, but different in size and proportion, and so related to A as to fulfill the following conditions:

19 For any time t on system A there will be a corresponding value of any one of the four characteristics above noted. Then for system B, let there be a time αt for which the value of the same characteristic will be related to the value for system A by a ratio β and where α and β are constant factors. This would mean that if curves for system B were run out similar to those for system A, then the two sets of curves would be geometrically similar, being related along the axis of abscissæ by the ratio α and along the axis of ordinates by the ratio β. It would mean, furthermore, that if the two sets of curves were plotted with suitable scale units, related for t in the relation of 1 to α and for any one of the four characteristics in the relation of 1 to the corresponding β, the two curves would become coincident. It will, of course, be understood that there will be one value of α for the time relation and two values of β, one for the three vertical dimensions or movements and one for the velocity v.

20 Now two systems A and B fulfilling the conditions thus indicated are said to have kinematic similitude.

21 It is then clear, if B, let us say, is a field case and A is of laboratory dimensions and if we can determine the coefficients α and β, that experimental observation on A will serve to determine through α and β the results to be anticipated for B.

22 We must next inquire as to the conditions for realizing such relation of kinematic similitude. Let us first assume that it *is* realized and by way of notation let:

p = ratio of conduit lengths	$= L$	ratio
q = ratio of the two values of c	$= c$	ratio
r = ratio of velocities	$= v$	ratio
s = ratio of the times	$= t$	ratio

23 Now referring to Equation [**1**], it has been stated (without present proof) that the expression $(L/g)(dv/dt)$ is the measure of an accelerating head; that is, the measure of the head required to produce the acceleration dv/dt. This head is, of course, a vertical dimension. Now since Equation [**1**] is a physical equation, it must

be homogeneous in all its terms. That is, all terms must represent vertical dimensions. In fact, it is readily seen that the right-hand side of the equation is composed of members, each one of which is a vertical dimension, and the algebraic combination of which is a measure of the difference between two water levels such as CD and C_1D_1, Fig. 1, and which (as previously stated) constitutes the accelerating head.

24 Now suppose two such equations, one relating to system A and one to system B. Then if the conditions of similitude are fulfilled there will subsist between the numerical values of the corresponding terms of these two equations, a constant ratio — that corresponding to vertical dimensions or vertical movements. This means specifically that the terms $(L/g)(dv/dt)$, H, y, cv^n, all represent vertical distances or movements, and that between them all, for systems A and B, there subsists the one constant ratio. If, therefore, we can determine this ratio for one, we have it for all. But with the notation assumed above, we have immediately:

$$v^n \text{ ratio} = r^n$$
$$cv^n \text{ ratio} = qr^n$$

25 Hence we shall have for all vertical dimensions or movements the ratio qr^n. We may denote this in general as the y ratio and have, therefore:

$$y \text{ ratio} = qr^n$$

26 Then since, in Equation [**1**], this ratio qr^n applies to each term individually, it must apply to the first term. Hence we may write:

$$\frac{L}{g}\frac{dv}{dt} \text{ ratio} = qr^n$$

27 But the L ratio $= p$, the dv ratio must equal the v ratio which is r, and the dt ratio must equal the time ratio which is s. Hence we have:

$$\frac{pr}{s} = qr^n$$

or

$$s = \frac{pr}{qr^n} = \frac{pr}{y \text{ ratio}} = \frac{p}{qr^{n-1}}$$

28 Again, in Equation [**2**], if the conditions of similitude are fulfilled, the ratio r will apply to each term individually. It must therefore apply to the first term and we may write:

$$\frac{F}{A}\frac{dy}{dt} \text{ ratio} = r$$

29 But the dy ratio must equal the y ratio and the dt ratio the t ratio. Hence we have:

$$\left(\frac{F}{A}\text{ ratio}\right)\left(\frac{qr^n}{s}\right) = r$$

or

$$\frac{F}{A}\text{ ratio} = \frac{rs}{y\text{ ratio}} = \frac{pr^2}{(y\text{ ratio})^2} = \frac{p}{q^2r^{2n-2}}$$

30 Now collecting these various ratios we have as follows:

$$L\text{ ratio} = p \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [3]$$

$$c\text{ ratio} = q \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [4]$$

$$v\text{ ratio} = r \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [5]$$

$$H,y\text{ ratio} = qr^n \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [6]$$

$$t\text{ ratio} = s = \frac{pr}{y\text{ ratio}} = \frac{p}{qr^{n-1}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad [7]$$

$$\frac{F}{A}\text{ ratio} = \frac{rs}{y\text{ ratio}} = \frac{pr^2}{(y\text{ ratio})^2} = \frac{p}{q^2r^{2n-2}} \quad . \quad . \quad . \quad . \quad . \quad [8]$$

and to which we may add:

$$F\text{ ratio} = (A\text{ ratio})\frac{pr^2}{(y\text{ ratio})^2}\text{ or }(A\text{ ratio})\frac{p}{q^2r^{2n-2}} \quad . \quad [9]$$

or

$$D\text{ ratio} = (d\text{ ratio})\frac{p^{1/2}r}{y\text{ ratio}}\text{ or }(d\text{ ratio})\frac{p^{1/2}}{qr^{n-1}} \quad . \quad . \quad [10]$$

$$V\text{ ratio} = (F\text{ ratio})(y\text{ ratio}) = (A\text{ ratio})\frac{pr^{2-n}}{q} \quad . \quad [11]$$

where D = diameter of surge chamber (assumed circular)
d = diameter of conduit line (assumed circular)
V = volume movement in surge chamber.

31 We have thus assumed the existence of various structural ratios, of a velocity ratio and of a time ratio, and have derived the relations among these ratios necessary in order that the conditions of homogeneity among the terms of any one equation may be realized.

32 Suppose now the two systems A and B set up in accordance with these various structural relations. It is then readily seen that we may choose initial conditions of operation which will fulfill the required relations throughout. That is, at the initial conditions, the requirements for kinematic similitude may be fulfilled by arbitrary adjustment. It is then readily shown that under these conditions the increments of y and v will likewise fulfill the relations for similitude, and hence the succession of values of y and v throughout the entire period of transition.

33 It thus results with two systems A and B, with structural relations as specified and for which the water movements are in accordance with Equations [**1**] and [**2**], that if the conditions of similitude are fulfilled at the beginning of the movements, they will likewise be fulfilled for subsequent times throughout the movement and thus the general conditions for similitude will be realized as assumed.

FORM OF RELATIONS FOR SIMILITUDE WHEN $n = 2$

34 It will next be of interest to note the form taken by these relations if the index n is taken equal to 2, as in the more common methods of treatment of the problem. Referring to [**6**]–[**11**], we shall have —

$$y \text{ ratio} = qr^2$$

$$\text{Time ratio } s = \frac{p}{qr}$$

$$\frac{F}{A} \text{ ratio} = \frac{p}{q^2r^2}$$

$$F \text{ ratio} = (A \text{ ratio}) \frac{p}{q^2r^2}$$

$$D \text{ ratio} = (d \text{ ratio}) \frac{p^{1/2}}{qr}$$

$$V \text{ ratio} = (A \text{ ratio}) \frac{p}{q}$$

SURGE CHAMBER WITH SPILLWAY

35 A case possessing both interest and importance is presented by a surge chamber fitted with an overflow weir or spillway at a certain fixed height.

36 Let B = Length of such spillway or weir
y_0 = height from datum to level of spillway edge
y = height from datum to surface of water
Then $y - y_0$ = depth of water on weir
In time dt let the water level rise dy
In time dt the flow along conduit line = $vAdt$
In time dt the discharge over the weir = $QB\ (y - y_0)^{3/2}dt$
In this formula for weir discharge Q is taken as containing all factors other than B and $(y - y_0)$

37 We have then —

$$vAdt = Fdy + QB\ (y - y_0)^{3/2}dt \quad . \quad . \quad . \quad . \quad . \quad [\mathbf{12}]$$

or

$$vA = \frac{Fdy}{dt} + QB\ (y - y_0)^{3/2} \quad . \quad . \quad . \quad . \quad . \quad . \quad [\mathbf{13}]$$

38 This equation together with [1] for the acceleration head will serve instead of [2] to determine the motion of the water level after the water reaches and rises above the weir edge.

39 To apply the principles of similitude to this case we must assume the weir coefficient Q the same for both weirs, field case and experimental set-up. We then note that each term of [13] is a quantity of the order of volume flow. Hence for similitude we must have the same ratio between the two systems A and B for each member of the equation. Thus we have:

$$vA \text{ ratio} = r\ (A \text{ ratio})$$

$$F\frac{dy}{dt} \text{ ratio} = F \text{ ratio} \times y \text{ ratio} \div t \text{ ratio}$$

or reducing,

$$F\frac{dy}{dt} \text{ ratio} = r\ (A \text{ ratio})$$

whence

$$[QB\ (y - y_0)^{3/2}] \text{ ratio} = r\ (A \text{ ratio})$$

and

$$B \text{ ratio} = \frac{r\ (A \text{ ratio})}{(y \text{ ratio})^{3/2}} = \frac{A \text{ ratio}}{q^{2}/y^{3n/2-1}} \quad . \ . \ . \ . \ [14]$$

40 If in this case we assume the index $n = 2$ we shall have —

$$B \text{ ratio} = \frac{A \text{ ratio}}{q^{3/2}r^{2}} = \frac{A \text{ ratio}}{q^{1/2}(y \text{ ratio})} \quad . \ . \ . \ . \ [15]$$

41 The volume of water discharged over the weir will be given by the expression —

$$V = QB \int (y - y_0)^{3/2}\, dt \quad . \ . \ . \ . \ . \ . \ [16]$$

We have then —

V ratio = product of individual ratios for expressions making up V as in [16] hence —

$$\text{V ratio} = \frac{A \text{ ratio}}{q^{3/2}r^{3/2n-1}}\, q^{3/2}r^{3n/2}\frac{p}{qr^{n-1}} = (A \text{ ratio})\ \frac{pr^{2-n}}{q}$$

and for $r = 2$,

$$V \text{ ratio} = (A \text{ ratio})\ \frac{p}{q}$$

42 This is seen to be the same as the volume ratio for change of volume in the surge chamber, as indeed it should be.

CASE WITH MULTIPLE SURGE CHAMBERS

43 The case with two surge chambers is indicated in Fig. 3 as a plan view. The main supply reservoir is at A, a small regulating reservoir at Q and a surge chamber of usual dimensions at P with the power plant lying beyond at R. With such an arrangement the reservoir Q functions as a large surge chamber, thus complicating the problem as affecting the movement of water in P.

44 With this combination of elements a disturbance from the conditions of steady flow involves, physically, the problem of three superimposed oscillatory systems; one by way of ABQ, one

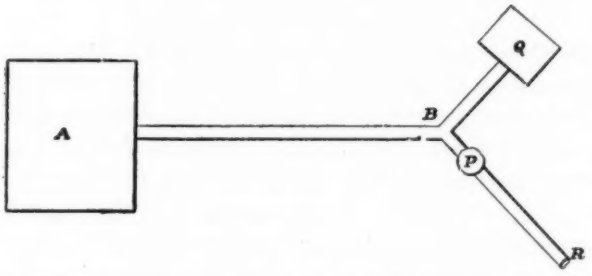


FIG. 3 DIAGRAM OF THE CASE OF TWO SURGE CHAMBERS

by way of ABP and one by way of PBQ. To define these conditions, six equations are required as follows:

$$\text{Motion in } AB,\ \frac{L_1}{g}\frac{dv_1}{dt} = H - z - c_1 v_1^n$$

$$\text{Motion in } BP,\ \frac{L_2}{g}\frac{dv_2}{dt} = z - y_2 - c_2 v_2^n$$

$$\text{Motion in } BQ,\ \frac{L_3}{g}\frac{dv_3}{dt} = z - y_3 - c_3 v_3^n$$

$$\text{Continuity at } B,\ A_1 v_1 - A_3 v_3 = A_2 v_2$$

$$\text{Continuity at } P,\ A_2 v_2 - F_2\frac{dy_2}{dt} = A_1 u$$

$$\text{Continuity at } Q,\ + F_3\frac{dy_3}{dt} = A_3 v_3.$$

where L_1, v_1, A_1, c_1 = respectively length, velocity, area and coefficient c for AB

L_2, v_2, A_2, c_2 = respectively length, velocity, area and coefficient c for BP

L_3, v_3, A_3, c_3 = respectively length, velocity, area and coefficient c for BQ

H = total static head from top of reservoir at A to datum at bottom of P

z = pressure head at junction B measured from datum at bottom of P

y_2 = level of water in P measured from datum at bottom of P

y_3 = level of water in Q measured from datum at bottom of P

u = velocity in AB correspondings to new demand in R.

45 In these equations as written, the positive directions of flow are taken as AB, BQ, BP.

46 Equations of this character and number are quite beyond the range of any practicable mode of mathematical attack, even by way of numerical integration. The methods by way of similitude hold, however, as with the simpler cases, and the model once set up, any condition of operation is readily examined. In setting up the model it is only necessary to use a uniform ratio of length and of diameter or area of conduit. These ratios together with the velocity ratio will then fix the ratio for all vertical distances, for time and for the cross-section area of surge chamber.

47 For a case with more than two chambers the same general principles hold, and any such case may be examined by experimental methods through the use of a model set up with ratios as above. Such cases, however, are not likely to be met with in practice.

48 For the case of Johnson's differential regulator [1] similar fundamental equations, with suitable interpretation, apply as for the same open chamber, and under these conditions the ratios and relations will be the same and the performance in detail may be investigated by means of the model method.

SPECIAL NOTES ON APPLICATION OF METHOD

49 In the application of these methods it should be noted that the assumption of a uniform value of the ratio q does not imply the assumption of the same values of friction coefficient for large and for small pipe. It simply implies that for the large pipe the velocity-friction head may be put in the form cv^n, the value of c being constant

[1] Trans. Am.Soc..C.E. vol. LXXVIII

over the range of velocity involved, and likewise that the similar head for the small pipe may be expressed in the same form, this value of c likewise being constant over the velocity range involved. The ratio between the two values of c will then be the constant q of the formulas.

50 Also it should be noted that in cases of design the c for the field case will usually be the subject of estimate based on judgment. This, however, is required for any computations or estimates regarding velocity of flow, capacity of line, power, etc., and the one estimate as to friction head will serve uniformly for these varied purposes. On the other hand, the value of c for the experimental model is a matter of direct measurement and in all cases should be so determined. A series of simple measurements of flow, time and drop in level between reservoir and surge chamber will serve as a basis for this determination for the model. Such observations plotted on logarithmic paper provide then a ready means for the determination of the values of c and n for the model, and this value of c compared with the assumed c (with the same n) for the field case will give the value of q as noted above.

51 *Surge Chamber Non-Uniform in Cross-Section Area.* It should be especially noted that with the experimental method, the surge chamber is not necessarily of uniform cross-section area. It may be tapering or of any form at will, so long as the model is made of corresponding dimensions. It should also be noted that the ratios of horizontal and of vertical dimensions are not in general the same. The result will therefore be geometrical similarity between the model and full-sized installation, but not the same proportions between horizontal and vertical dimensions.

52 It is simply necessary that, at corresponding vertical dimensions as determined by one ratio, the horizontal dimensions are also similar, as determined by the other ratio. This is further illustrated by Fig. 4 showing the relative proportions for a full-sized and a model surge chamber, the former as installed on the line of the Los Angeles Aqueduct Power System.

53 *Governor Action.* At an earlier point in the paper, reference has been made to the various assumptions which may be made regarding the results of governor action during the period of change and to the fact that no assumption which can be made in precise terms will represent the real program in an actual case. Due to the relative simplicity of the equations resulting, however, assumption (a) was implied in deriving the basic Equations [**1**] and [**2**].

54 It should again be especially noted that no matter which of these assumptions may be made, the ratios for similitude remain unchanged. Equations of the type of [1] remain the same, while equations of the type of [2] remain the same in form with the substitution for v_2 of a velocity u in AB, suited to the particular assumptions made.

55 With (*a*), v_2 as already noted is the velocity in AB necessary to give, under final steady-flow conditions, the power required under the changed conditions.

56 With (*b*), for v_2 we must substitute a velocity u defined as the velocity in AB which would bring the water required to meet the new power requirements under the momentarily changing values of available head and the efficiency of the wheel.

57 With (*c*), the velocity u is defined the same as in (*b*), with

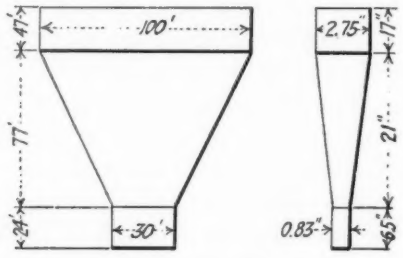


FIG. 4 RELATIVE PROPORTIONS FOR A FULL-SIZED AND MODEL SURGE CHAMBER

the omission of the influence due to variation in efficiency of wheel.

58 With (*d*), the velocity u is defined as that corresponding to the total flow through the valve, set as specified, and with the momentarily changing value of the available head as affected by the changing level in the surge chamber.

59 In all of these cases the term v_2 or u is a velocity and the ratios for similitude remain the same, being unaffected by these various assumptions.

60 It therefore becomes a question of interest as to which of these four assumptions can be most easily realized in the manipulation of a model.

61 It is readily seen that assumption (*d*) most readily meets the requirements. For any specified power development it is a simple matter to determine the volume flow required under the head corresponding to such flow. To realize the corresponding condition with

the model requires simply the setting of a control valve at P, Fig. 1, in accordance with a previously determined calibration.

62 On the other hand, if any of the other assumptions is to be fulfilled by model, the rates of volume flow corresponding to the special conditions must be determined and the setting of the control valve determined therefrom, taking into account the height of the water in the surge chamber. The valve setting will thus depend on such height and hence the valve will require adjustment continuously during the period of transition, corresponding to the rise and fall of the water in the surge chamber.

63 As a manipulative program it is entirely possible to realize this through the use of a cam form of control, the movement of the cam being determined in step with the movement of the water in the chamber. Such a program of control is, however, somewhat complex in character and adds in marked degree to the time required for making and assembling the model equipment.

64 In order, therefore, to realize in the highest degree the advantages of the model method, it is desirable to accept assumption (*d*) as representing with a sufficient degree of approximation the operative conditions of the plant. An examination of the conditions of operation of a power plant will serve to show, furthermore, that with supposition (*d*) as compared with either (*a*), (*b*) or (*c*), the resulting conditions as regards surge are more severe and hence that the error will be on the side of safety. It is also evident that the influence of the variation of level of the water in the surge chamber on the general program of volume discharge or of power developed at the power house will be relatively small as the range of such fluctuation is small compared with the difference in level between water in surge chamber and power house and relatively large in the inverse case.

65 The whole question of the use of a model for the study of such problems turns therefore largely on the degree of closeness to which the manipulation of the model may be made to represent the proposed programs of change in the field installation, and more specifically in the actual power plant; or again on the extent to which the simple model program as indicated in assumption (*d*) may be accepted as satisfactory for the purposes in view. As indicated above, it appears that, in all cases where the range of fluctuation of the water level in the surge chamber is small compared with the difference in level between surge chamber and power house, the simple program of (*d*) may be accepted as satisfactory for all practical purposes. If such fluctuation should, on the contrary, be large,

the use of the model would presumably require such further devices as would make possible a closer approximation of the program of water flow in the model outlet to the actual or proposed program in the power house.

66 A further point should be here noted, and that is that the model proper in its correspondence with the field case covers only the main supply line and the surge chamber. It does not necessarily extend to the line from the surge chamber to power house. In the model the line from surge chamber to discharge valve may be considered simply as a means for realizing a desired discharge through the setting of a calibrated valve. From this view, therefore, it is simply necessary to have such size of line and such head as shall insure, through the discharge valve, the rates of flow covering the ranges contemplated. At the same time it is generally desirable to give to this discharge line a vertical dimension not widely dissimilar from that corresponding to the field case, in order that the range of fluctuation of the level in the surge chamber may bear to the difference in level between the surge chamber and power plant approximately the same ratio in both model and field cases.

67 *Special Programs of Change.* In addition to the usual problem as presented by a sudden change in power demand followed by substantially uniform conditions, the model method readily serves for the examination of two types of problem, neither of which is practically susceptible of treatment through computation methods. These are as follows:

68 *a* Required the cumulative result of periodic changes of any specified magnitude and with any specified frequency. Thus for illustration, we might have a proposed program of additions to the load, each of 10 per cent total load, and with specified intervals. If the latter should be such as to bring the load changes into approximate synchronism with the surge chamber movements, the cumulative results might become very serious; while otherwise the resultant movement would be relatively small. Again, we might have the condition of a surging load, alternate increase and decrease and with any proposed frequency. This again, if in approximate synchronism with the surge-chamber movements, may result in the most extreme and serious conditions in the latter.

69 All such problems are most readily examined by the model. It becomes necessary simply to determine the change of flow corresponding to the proposed change in power and to note the corresponding settings of the control valve. Then with the corre-

sponding time interval known, it is simply a matter of manipulative procedure of the control valve, with corresponding note of the resultant movement of the water in the surge chamber. In this general manner the results of all forms of varied or periodic programs of change in power demand, or of flow in general, are readily examined.

70 *b* Required the time over which a specified power change should be extended in order that the resultant surge-chamber movement may not exceed a specified amount. Thus with a proposed size of surge chamber in a given case, a sudden change of 80 per cent or of 100 per cent of the load, for example, will perhaps produce a surge-chamber movement resulting in extreme overflow at top, or in the indraft of air at the bottom. In either case it may be desired to determine the necessary duration of a specified load change in order that the surge-chamber movement may not exceed specified limits. As a manipulative program this calls simply for the movement of the control valve between specified stops gradually at an approximately uniform rate and involving such total time interval as will meet the requirements within the surge chamber. Experience of the writer shows that this is readily realized by successive trials, and that, as a problem in manipulation, it presents no serious difficulties whatever. The time interval for the model, thus determined, is then to be multiplied by the time ratio, thus giving the required time for the field installation.

71 *Experimental Detail.* The experimental program, in general, connected with an investigation of this character is simple and may safely be left to the initiative of the interested reader. A few suggestions on particular points may, however, be acceptable.

72 The control valve is preferably of the cone or needle type with long taper, so that a considerable stem movement will be required between closed and full open. The stem may be controlled by hand lever or otherwise as most convenient, with, in any case, an index moving over a graduated scale. The valve may then be calibrated for steady-motion discharge or velocity by simultaneous observations of valve setting and weight of water discharged in a given time. Observation of the height of water in the surge chamber for each of these settings furnishes likewise data for a series of values of the friction-velocity head, cv^n, for the model. A carefully drawn curve between discharge and cv^n will then furnish a ready means of determining discharge by a reading of water level in the surge chamber. From this point on, the valve settings should be used simply to realize approximately the conditions desired, the actual velocity

or discharge being taken as that corresponding to the water level in the surge chamber under steady flow conditions.

73 For reading the movement of the water in the surge chamber, either a float with stem or a gage glass on the side may be employed. With the latter there is time lag and some correction may be required. The author has found the former method the preferable. The stem may be furnished with an index moving in front of a suitable scale and with all usual proportions the movement is slow enough to readily permit the reading of maximum and minimum points.

ILLUSTRATIVE PROBLEMS

75 I. Suppose an actual case characterized by the following data:

Length of conduit, ft.	20,000
Diameter of conduit (assumed circular), ft.	10
Upper velocity, ft. per sec.	10
Friction-velocity head at 10 ft. per sec. velocity (assumed), ft.	50
Proposed diameter of surge chamber, ft.	36

76 Suppose now that we propose to use for the model conduit 50 ft. of pipe 2 in. in diameter and that we select arbitrarily a velocity ratio of 2.5. Suppose further that with this pipe set up we find experimentally $n = 1.85$ and $c = 0.1695$.

77 Then assuming the same value of n, we find for the field case $c = 0.706$. We have then —

$p = 20{,}000 \div 50$	$= 400$
$q = 0.706 \div 0.1695$	$= 4.17$
r	$= 2.5$
r^n	$= 5.45$
y ratio	$= 22.72$
t ratio	$= 44.0$
D ratio	$= 132.0$
$D = 432 \div 132$	$= 3.27$ in.

78 If then we set up the model in accordance with these dimensions, we have only to carry out the program of velocity change corresponding to the proposed field program, observe vertical movements and times, multiply the former by 22.72 and the latter by 44.0 and we shall have the vertical movements and times to be anticipated in the field case.

79 II. Suppose that it is proposed to fit the actual chamber with a spillway 10 ft. wide with edge 6 ft. above static level in supply reservoir. What will be the width and location of the model spillway?

80 The edge will be located at a height above static level in the supply reservoir measured by $6 \times 12 \div 22.72 = 3.17$ in. The width ratio from [**14**] is 82.9, hence the width is $10 \times 12 \div 82.9 = 1.45$ in.

81 III. As a further example drawn from actual practice, reference may be made to Fig. 2 with dimensions as follows:

$AB = 14{,}000\,f$ $\quad d_1 = 11.25\,f$ $\quad d_3 = 11.25\,f$
$BQ = 1{,}191\,f$ $\quad d_2 = 10.10\,f$ $\quad D = 35.00\,f$
$BP = 137\,f$

where d, d_2, and d_3 refer respectively to AB, BQ and BP.

82 The area of Q varied with elevation from 120,000 to 150,000 sq. ft., constituting a small regulating reservoir. The following proportions and dimensions were taken for the model:

$p = 240$ $\quad$ also $d_1 = 1.5$ in.
hence $AB = 58.23\,f$ $\quad d_2 = 1.342$ in.
$BQ = 4.96\,f$ $\quad d_3 = 1.5$ in.
$BP = 0.57\,f$ $\quad r = 3$

83 By experiment c and n for the model were found to be 0.195 and 1.823.

84 By estimate, for the field case, c with the same value of n was taken as 0.421.

85 This gives $q = 2.16$ and y ratio $= q^n = 16.00$.

86 We then find from the proper formulas —

t ratio $= 45$
F ratio $= 68{,}200$

87 This gives F for model = 2.04 sq. in. which corresponds to a circular chamber of diameter 1.60 in.

88 The same F ratio determines also the model reservoir. This was made as a wooden box with sloping sides, so adjusted as to give, over the possible range of change of level, the proper values of surface area.

89 The pipe of equivalent diameter 1.342 in. was made by filling in a segment of a 1.5-in. pipe to a point giving the proper area.

90 This combination of elements, with a suitable discharge valve, completed the set-up of the model. The investigation made possible by this model covered the entire range of flow from closure up to the full flow of 1000 sec-ft. and in both directions, demanded

and rejected load. These results are shown graphically in Figs. 5 and 6. Fig. 5 relates to the combination of surge chamber and reservoir, while Fig. 6 refers to the surge chamber in operation alone, the reservoir being shut off.

91 For the conditions of Fig. 5 the diagram shows the maximum water movement (for the field installation) for the first swing following any sudden change of load in either direction. The vertical scale gives elevation in feet, and the horizontal scale flow of water in second-feet.

92 The curve *A-B* shows the level of water for steady

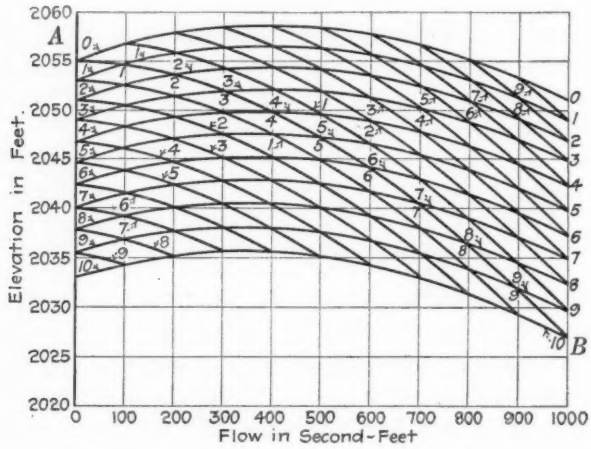


FIG. 5 FLOW TO BE ANTICIPATED WITH SUDDEN CHANGE IN EITHER DIRECTION FROM ANY INITIAL LOAD V_1 TO ANY FINAL LOAD V_2

(Surge chamber in combination with auxilliary reservoir.)

conditions for flow varying from 0 to 1000 sec-ft. The diagram shows two sets of curves:

1 Curves running clear across the sheet from left to right and crossing the curve *A-B*

2 Curves approximately parallel to the curve *A-B* and trending, therefore, downward to the right.

93 Any problem involves two values of the flow — the initial flow V_1 and the final V_2.

94 To determine the surge movement for demanded load, go to point V_1 on the curve *A-B* and drop vertically to that curve of set 1 which cuts the curve *A-B* at V_2. The point thus indicated

will give the water level for the first surge. If V_2 falls between the curves as plotted, the interpolated value is easily read.

95 Or again, with the plant in operation under flow V_1, let there be a sudden demand for an *additional* flow V. In this case we may find $V_2 = V_1 + V$ and proceed as above, or otherwise we may

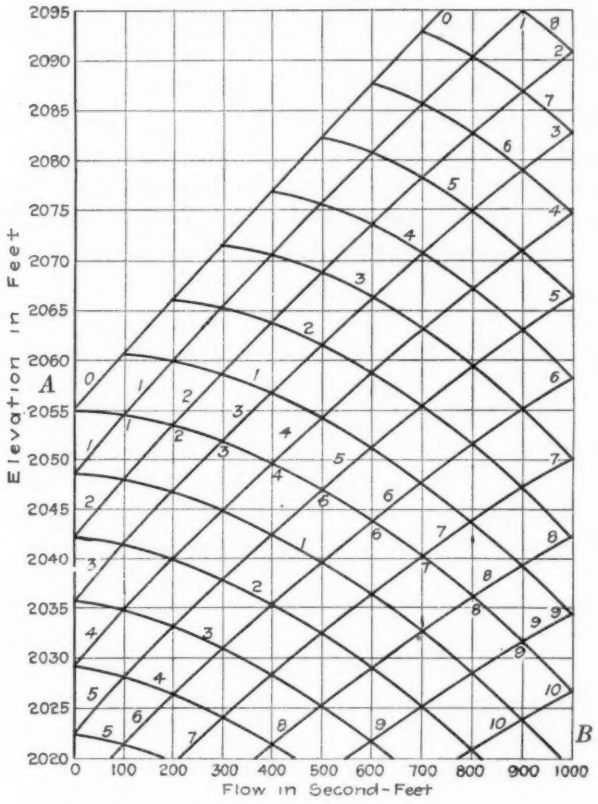


FIG. 6 FLOW TO BE ANTICIPATED WITH SUDDEN CHANGE IN EITHER DIRECTION FROM ANY INITIAL LOAD V_1 TO ANY FINAL LOAD V_2

(Surge chamber alone.)

go to V_1 on the curve A-B and then drop vertically to that curve of set 2 corresponding to the additional flow V. The point thus indicated will give the extreme level reached.

96 In the case of rejected load, the operation is entirely similar except that the curves lying to the right and above the curve A-B are to be employed.

97 In Fig. 6 for the surge chamber alone, the manner of plotting and method of use are the same as for Fig. 5.

98 These diagrams thus give graphically the results to be anticipated with a sudden change in either direction from any initial load V_1 to any final load V_2 and for the surge chamber either alone or in combination with the auxiliary reservoir. The influence of the latter in reducing the extent of the surge is thus brought out in a striking manner.

99 It is perhaps unnecessary to add that the numerical work required to derive corresponding results from the six different equations applying to this case would be entirely prohibitive from a practical viewpoint, and if any investigation is to be made of a case of this character, the experimental model method, using the law of kinematic similitude, as herein set forth, seems to be the only recourse available.

No. 1828

THE MOODY EJECTOR TURBINE

By S. Logan Kerr, Philadelphia, Pa.
Junior Member of the Society

ONE of the greatest difficulties met in the operation of low-head hydroelectric power plants is the reduction of capacity during flood periods. It is usually impossible, owing to the nature of the surrounding country and the enormous quantity of water required, to provide storage reservoirs to such an extent that all excess water flowing during these flood periods can be entrained and used when the stream flow is below normal. Some regulation, of course, is effected in a few instances, but for the most part there is sufficient only to carry the plant over the daily peak.

2 Where the storage is thus limited, owing to the character of the country surrounding the plant, the flow of water during the flood season is much greater than is required by the turbines. This excess is therefore wasted over the dam, and in addition causes the tail water to rise in level.

3 Since the turbine operates upon the difference in level of the head and tail waters, should these levels vary the same amount, then this difference or head would be constant. Usually this is not the case, for with a given rise in level of the water going over the dam, the rise in the level of the tail water is greater, thus decreasing the net head. This effect is illustrated in Fig. 1. The head at low water is evidently much greater than at high water as reference to this sketch will show. The line intermediate between the two extremes shows the effect when the level of the head water is increased by a small amount. It is evident even in this case that the net head is decreased.

4 Since the horsepower of the turbine varies as the three-halves power of the head acting upon it, the output of the station is

Extracts from a thesis submitted to the University of Pennsylvania for the degree of Bachelor of Science in Mechanical Engineering by the author in June 1921. Awarded A.S.M.E. Junior Prize for the best paper during the year 1921.

decreased in the same ratio, but as the operation is of necessity at constant speed, the efficiency may be less as well, depending upon the characteristics of the wheel, causing a still further decrease in capacity. Thus the resultant loss in power may be very great, depending upon the rate of off-flow of the tail water, which in turn is dependent upon the character of the stream bed below the dam.

5 As was stated above, in low-head plants where the quantity of water required to generate a given amount of power is large, the storage facilities are usually small. Therefore the turbine should be designed to give maximum efficiency at rated output when the head is a maximum, for at this time water is scarce and economical operation is imperative.

6 Due to their inherent characteristics, the high- or relatively

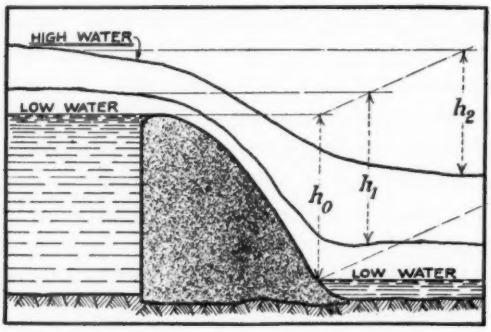


FIG. 1 SKETCH SHOWING DECREASE IN EFFECTIVE HEAD DURING PERIODS OF HIGH WATER

high-speed wheels used in low-head work give as a rule a rather narrow operating range at maximum efficiency, and a small overload capacity beyond this point of maximum efficiency. To design a turbine, then, that would deliver the required output under a reduced head would be inadvisable, for this same turbine would not operate at best efficiency when delivering rated horsepower under the higher normal head resulting from low water.

7 This is illustrated in Fig. 2 where the turbine is designed to deliver 100 per cent of rating at 75 per cent of normal head. When this same turbine is operated under 100 per cent head, its capacity is increased to 140 per cent of rating; but as only 100 per cent is required, it is evident that the efficiency will be several per cent less than the maximum. This method of meeting the problem would be

wasteful from the standpoint of economy of operation at maximum-head conditions.

8 Another method would be to install a number of additional units that could be placed in operation when the capacity of the plant was cut down at high water. Such a plant would require a much greater expenditure of money in the first cost, which could render a return on the investment for a short period only each year. This, then, is a possible solution, but the resultant fixed charges upon the plant would be so great, in many cases, as to make the cost of power prohibitive.

9 There remains this problem: to design a turbine that will

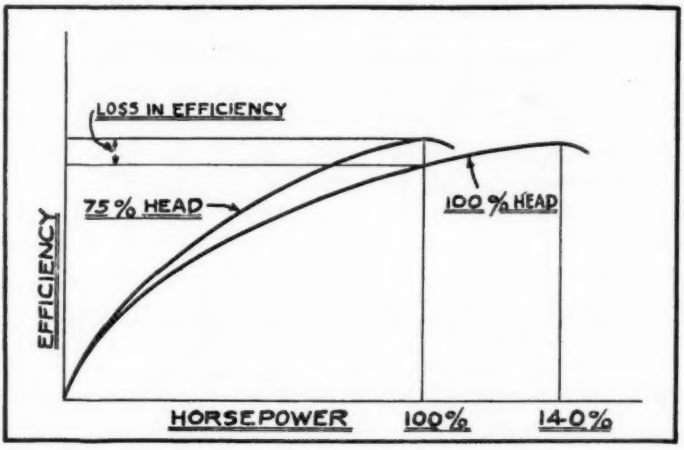


FIG. 2 SKETCH SHOWING DISADVANTAGE OF DESIGNING TURBINE FOR MAXIMUM EFFICIENCY AT THE MINIMUM HEAD CONDITIONS

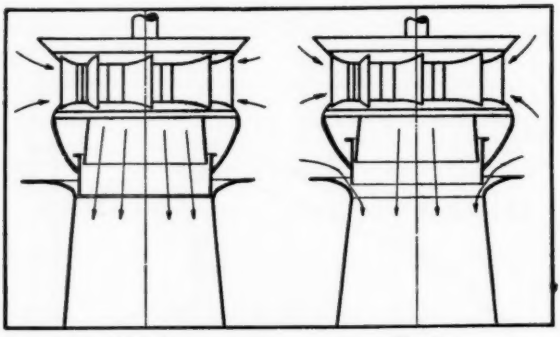
operate as economically as possible at times when the available supply of water is small, and yet develop normal rated power when the overabundance of water decreases the effective head.

THE MOODY EJECTOR TURBINE

10 A turbine designed to meet these requirements has been developed by Mr. Lewis F. Moody. The apparatus consists of a turbine of the normal type designed to operate at maximum efficiency under the average head prevailing at times of low water, but at the top of the draft tube, just below the runner, an annular opening is made in the wall of the tube, the amount of this opening being controlled by a cylinder gate.

11 When the turbine is operating normally, this gate is closed and the contour of the draft tube is unchanged, as shown in Fig. 3. When the head falls off due to the rise of the tail water, this gate is opened slightly, allowing a jet of water at high velocity to enter the draft tube without first passing through the runner. This case is shown in Fig. 4. The action of the jet, expressed simply, is to "pull" more water through the runner and hence to increase the horsepower output of the turbine.

12 The jet actually creates an additional suction head that compensates for the loss in the difference in level between the head and tail water so that the turbine is in reality operating under a



FIGS. 3 AND 4 SKETCHES RESPECTIVELY SHOWING THE EJECTOR CLOSED AND OPEN

head made up of two parts: first, the effective head measured as the difference in levels of the head and tail water (less, of course, the frictional head in the water passages); and second, the "suction head" produced by the action of the jet of water admitted below the runner. For this reason the device has been called an "ejector turbine."

TESTS MADE ON THE TURBINE

13 The apparatus described above was set up and tested in the I. P. Morris Experimental Laboratory of the William Cramp & Sons Ship & Engine Building Company in Philadelphia. The turbine was equipped with guide vanes placed in the ejector. The function of these vanes, which will be called ejector vanes hereafter in order to avoid confusing them with the guide vanes and

speed ring vanes of the turbine proper, is to give the water entering the ejector an initial whirl, so that there will be a minimum amount of interference with the water coming off the runner.

14 In the first series of tests (A1 to A6) model runner No. 55 was used. This runner (16 in. in diameter at the throat) is of the mixed-flow type and has a specific speed of 84.4 at best ϕ. The symbol ϕ represents the coefficient of peripheral velocity or the ratio of the linear velocity of a point on the runner to the theoretical spouting velocity of the water under the operating head. This ratio may be based on any one of several diameters, but in this article, the ϕ based on the diameter at the throat of the runner is meant. The values of specific speed (N_s) are all given in the English foot-pound system.

15 The first test (A1) was made at the maximum opening of the ejector (3.155 in.). The measurement of these openings was made by taking the average of a series of readings scaled off at various points around the ejector. A series of ϕ-Efficiency and ϕ-Horsepower curves under 1 ft. head and 1 ft. throat diameter was drawn for each test and the performance curves calculated from these. Description of the necessary calculations and of the method employed in testing are given in appendices to the complete paper.

16 A series of tests was made, each with a decreasing opening of the ejector until a point of 0.824 in. was reached. The next test (A6) was made with the ejector gate entirely closed.

17 However, this test did not give as good efficiency as was previously secured with the same runner and draft tube in another test. Upon investigation, it was found that the cylinder gate of the ejector did not seat properly upon the lower ring and that a small opening was left on one side of the turbine, thus admitting a small amount of water on about one-third of the periphery of the ejector. It is of interest to note that for any given gate opening of the turbine, although the efficiency was reduced about five per cent from that previously obtained, there resulted a slight increase in horsepower with even this small ejector opening.

18 In plotting performance curves of the values in test No. A6 the test efficiencies were stepped up to correspond with those obtained in previous tests. This is permissible, for in tests Nos. B6 and C2 made with runner No. 65, the ejector was calked tight to eliminate this difficulty and to correspond to the more perfect construction that would be found in large units built for commercial purposes. The results in these tests gave efficiencies that checked

very closely with those obtained when the same runner was tested without the ejector installed.

19 Another series of tests was made using runner No. 65 in place of runner No. 55. This runner (16.25 in. in diameter at the throat) is of the "diagonal-flow" type and has a specific speed of approximately 100. This series of tests (B1 to B6) was run with varying openings of the ejector, but the angle of the ejector vanes was changed to conform with the variation in the theoretical value of the whirling component of the water leaving the runner with the increase in specific speed.

20 When this series had been run, it was thought advisable to test runner No. 65 with the ejector vanes removed in order to study the effect of the ejector water without this initial whirl.

INVESTIGATION OF THE INFLUENCE OF EJECTOR UPON THE FLOW IN DRAFT TUBE

21 It was desired as well to see the effect that the water injected into the draft tube through the ejector would have upon the normal flow of water coming off the runner. In order to measure the velocity of water in the draft tube it is necessary to know the direction that it takes, since the flow is not the same as in a pipe line or canal, but moves with a whirling motion and follows a path similar to a helix in form. The steepness of this helix depends upon a great number of conditions, principal among which are the type of runner, the speed, and the load under which the turbine is operating.

22 An instrument was developed by the writer with the cooperation of the staff in charge of experimental work of the I. P. Morris Department especially for this test to measure the velocity and direction of flow in the draft tube. The apparatus, because of its form and the use for which it was designed, has been called a "pitot velometer."

23 The pitot velometer consists of a double pitot tube similar to the "Pitometer" with symmetrical openings 180 deg. apart. The head was arranged so the maximum dimension would pass through a hole the size of the outside diameter of standard $\frac{3}{8}$-in. pipe. The whole head, rod, and end connections are free to rotate and move in and out from center line of the draft tube. A pointer is fixed on the outer end of the instrument, and a dial hung on a ball bearing and weighted to keep the zero of the scale vertical showed the position of the center line of the openings in the head.

24 These two openings were attached to the legs of a differential gage filled with mercury and when the head was turned to a position so that the two openings were in a line with the flow of the water, the velocity head was indicated on the gage. When the rod was rotated through 90 deg., the center line through the openings was perpendicular to the flow, and the two legs of the gage stood at the same level. It was found that the apparatus was very much more sensitive in this position than with the openings in line with the flow, and that the angle of deviation from the vertical could be more accurately read.

25 For this reason two pointers were attached to the rod;

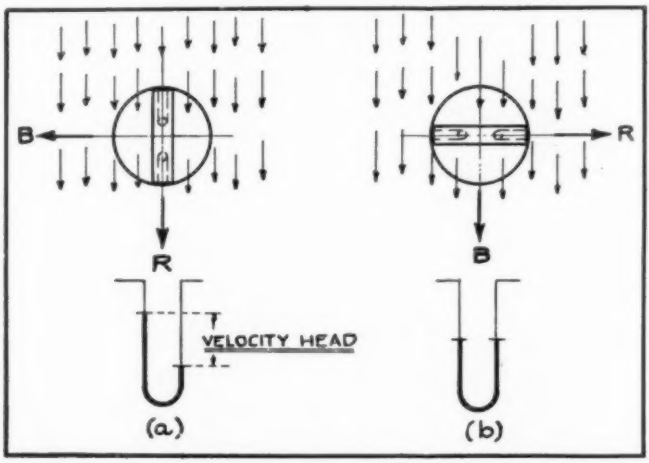


FIG. 5 POSITIONS OF PITOT VELOMETER (*a*) WHEN READING VELOCITY AND (*b*) WHEN READING DEVIATION FROM VERTICAL

(R indicates red pointer; B, black pointer.)

one painted black was set at right angles to the line through the center of the openings, and the other painted red set parallel with this line.

26 In taking a reading, the rod was set at the desired point on the diameter of the draft tube and rotated until the two legs of the differential gage stood equal; this position is shown in Fig. 5 (*b*). The reading of the deviation was recorded as shown by the black pointer. The red pointer was then brought to this point, automatically placing the openings in line with the flow and the

velocity read by the difference in levels of the mercury columns. This position is shown in Fig. 5 (*a*). The scale of the dial was graduated in degrees and half-degrees. It was possible to measure the deviation of this velocity from the vertical accurately to within a single degree.

27 The scale of the mercury gage was graduated to read zero when the two legs were equal; below this line the scale was laid off in inches of mercury. Above the zero line the scale was arranged to give the velocity directly in feet per second. The pitot velometer was located approximately 24 inches below the horizontal center line of the turbine.

28 The diameter of the draft tube at the point where the instrument was installed was $19\frac{3}{4}$ in., but as the flow was investigated on one side only of the tube, the horizontal motion of the head was one-half of the diameter or $9\frac{7}{8}$ in.

29 Readings were taken at each even inch from the vertical center line up to and including a point at a radius of 9 in., thus making a total of ten readings for one complete traverse.

30 It was thought best, due to the time required (seven to ten minutes) for one complete traverse of the draft tube, with ten readings each of velocity and angle of deviation from the vertical, to take traverses only at points within the possible operating range of the turbine, or of a turbine of this type installed in a power plant.

31 Since the limits of maximum and minimum ϕ for this range could easily be determined, three traverses were made for each gate opening at points within these limits. A large number of traverses were made, two examples of which are shown in Figs. 14 and 15.

32 Two tests were made, the first (C1) was run with the ejector open about one and one-half inches, the second (C2) was made with the ejector closed and calked, to have, as a basis of comparison, the flow of water coming from the runner uninfluenced by the ejector water.

RESULTS OF TESTS

33 In order to show the increase in horsepower obtained by use of the ejector over that normally delivered by the turbine, a series of performance curves was drawn with efficiencies as ordinates and horsepower under 1 ft. head reduced to 1 ft. throat diameter as abscissae. These curves were drawn for each of the tests in the first and second series. The envelope of these curves was then

drawn in and is taken as the performance of the turbine and ejector combined.

34 It may be seen from Figs. 6 and 9 that a considerable in-

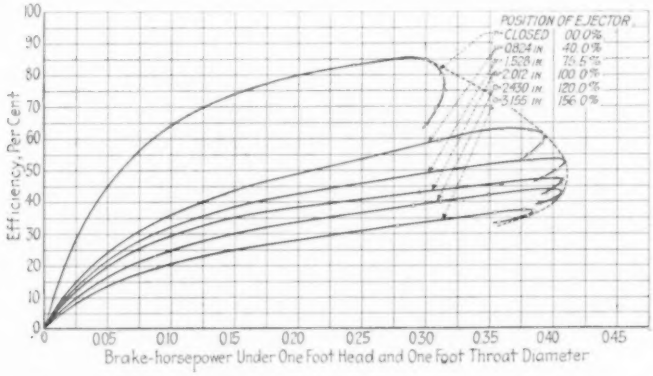


FIG. 6 CURVES SHOWING OPERATION AT NORMAL ϕ WITH VARYING EJECTOR OPENINGS AND EJECTOR VANES IN PLACE; RUNNER NO. 55

crease over the rated output is obtained by the use of the ejector. This increase is 36.7 per cent based on rating of 0.300 hp. under 1 ft. head and 1 ft. throat diameter for runner No. 55 and 30.0 per cent

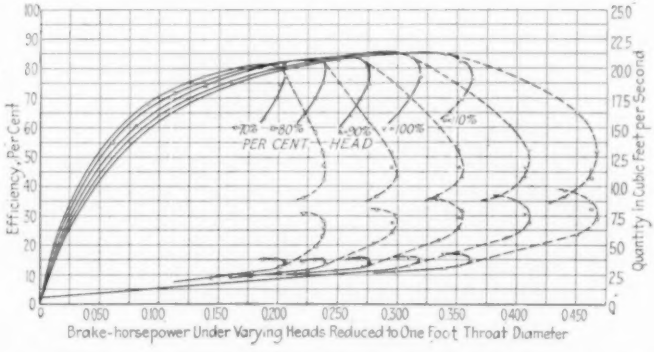


FIG. 7 CURVES OF HORSEPOWER, EFFICIENCY AND QUANTITY FOR RUNNER NO. 55 UNDER VARYING HEADS

Normal Operation Indicated by Solid Lines and Operation with Ejector by Broken Lines.

for runner No. 65 equipped with vanes in the ejector. Based on the maximum power obtainable these values are 34.5 per cent for runner No. 55 and 17.0 per cent for runner No. 65.

35 A second set of curves (Figs. 7 and 10) was plotted showing

the performance of the turbine and ejector under varying heads, each curve consisting of two parts, the normal performance curve and the envelope of the several ejector performance curves. Curves

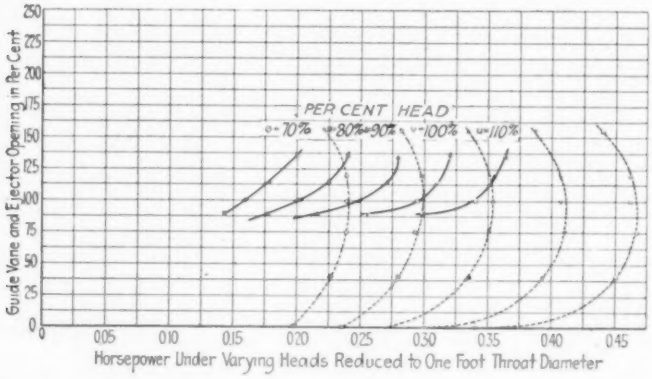


FIG. 8 CURVES OF HORSEPOWER FOR VARYING HEADS FOR DIFFERENT GATE (SOLID LINES) AND EJECTOR (BROKEN) OPENINGS, RUNNER NO. 55.

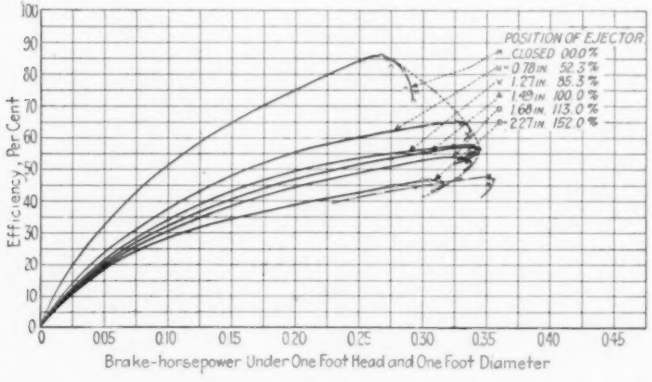


FIG. 9 CURVES SHOWING OPERATION AT NORMAL ϕ WITH VARYING EJECTOR OPENINGS AND EJECTOR VANES IN PLACE; RUNNER NO. 65; ONE CURVE WITH VANES REMOVED

are plotted for 70, 80, 90, 100 and 110 per cent of normal head, the latter being taken as 1 ft. to have a standard for a basis of comparison. Along with these curves have also been plotted curves of horsepower against quantity.

36 Guide vane and ejector openings have also been plotted

against horsepower and appear as Figs. 8 and 11. In all of these results there has been no allowance made for the increased frictional losses in the water passages leading up to the turbine, since the

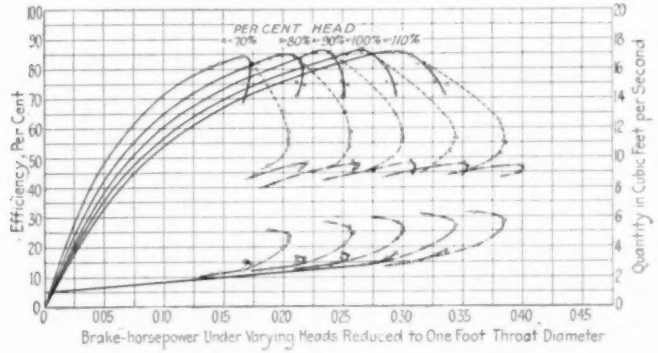


FIG. 10 CURVES OF HORSEPOWER, EFFICIENCY AND QUANTITY FOR RUNNER NO. 65 UNDER VARYING HEADS

Normal Operation Indicated by Solid Lines and Operation with Ejector by Broken Lines.

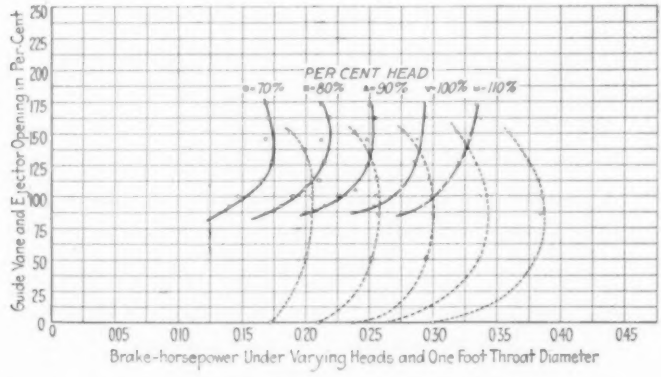


FIG. 11 CURVES OF HORSEPOWER FOR VARYING HEADS FOR DIFFERENT GATE (SOLID LINES) AND EJECTOR (BROKEN LINES) OPENINGS, RUNNER NO. 65

model tested was of the open-flume type. Therefore the losses resulting from the increase in quantity flowing must be calculated for each individual case where the ejector is to be applied. It should be noted that the results include some of the losses, namely, the losses within the turbine and the losses in the draft tube, but not

the losses that would occur in the penstock or the casing. It is probable that these will not be excessively large, since low-head installations have relatively short penstocks which are usually integral with the dam.

37 It may be noted that with runner No. 55, normal rated

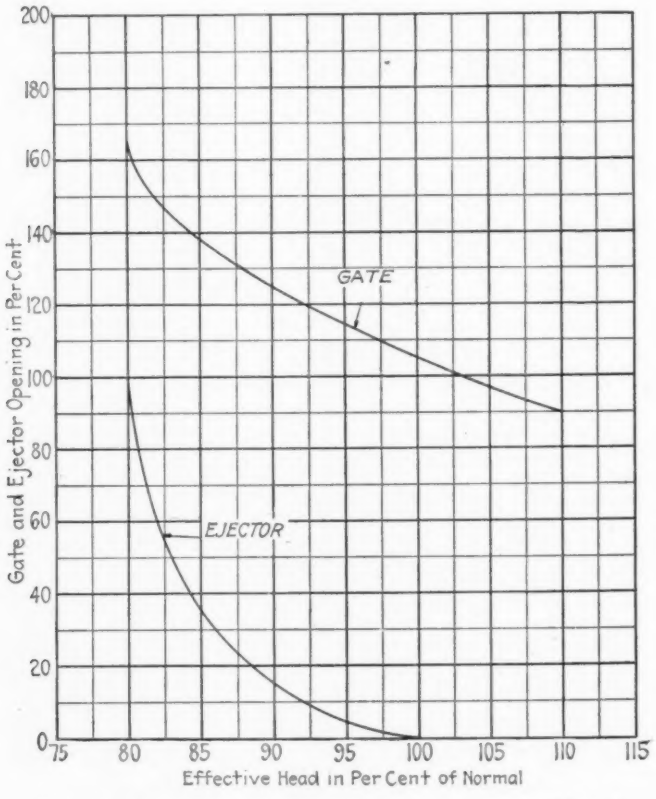


FIG. 12 CONSTANT-HORSEPOWER CURVES, RUNNER NO. 55

horsepower can be maintained under 80 per cent head and under 83 per cent head with runner No. 65, when both are equipped with guide vanes in the ejector.

38 From the curves of horsepower, gate and ejector opening, another series of curves was obtained (Figs. 12 and 13). These might be called very appropriately "Constant-Horsepower Curves." The ordinates are per cent ejector or gate opening and the abscis-

sae are per cent normal head. If, therefore, the per cent of normal head under which the turbine is operating is known, by finding the points where the gate and ejector opening curves cross this line, the proper setting of the turbine and ejector gates that will maintain rated capacity may be read. These curves are plotted for each of the two main series of tests.

39 In Fig. 10 (curves of horsepower and efficiency under vary-

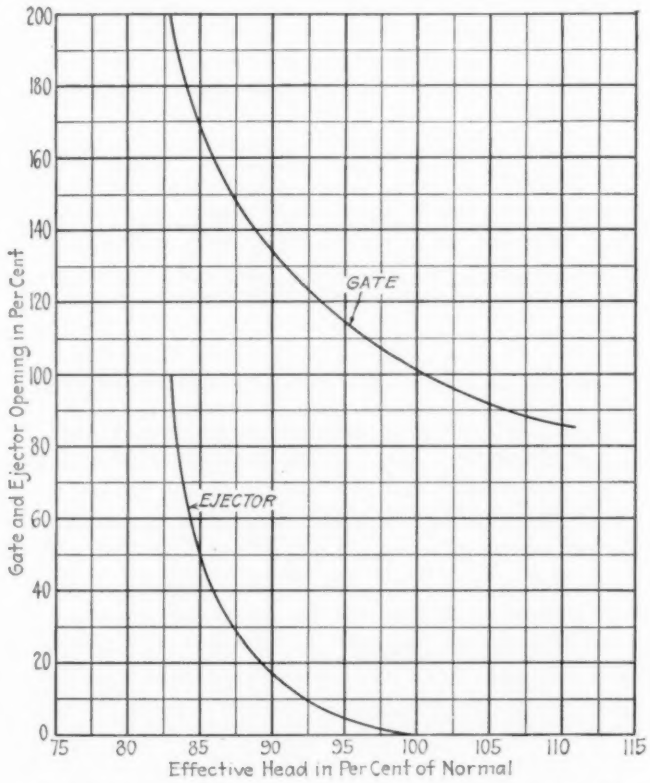


FIG. 13 CONSTANT-HORSEPOWER CURVES, RUNNER NO. 65

ing heads for runner No. 65) a set of small curves, one for each 10 per cent of normal head from 70 to 110 per cent, are drawn for comparison with the other performance curves. These are the results of test No. C1 (runner No. 65 with ejector vanes removed), and it may be seen that there is a slight numerical increase in horsepower over that obtained with the vanes in place, but judging from the

position of these curves with relation to the others and to the shape of the envelope of the curves of varying ejector openings, it is reasonable to suppose that the maximum increase obtainable with the ejector vanes removed is much greater than the actual numerical difference would indicate. It would seem that the turbine was operating on the drowned part of the envelope.

40 The explanation of this is probably that the increase in area of the opening with the vanes removed was greater than that required to secure maximum horsepower, and had the effect of drowning the turbine. If the opening had been smaller, the actual maximum horsepower obtainable with the vanes removed would have been secured.

41 In all the foregoing tests the turbine was equipped with the Moody spreading draft tube, and it is reasonable to suppose that the results are considerably better than could have been obtained with a straight or curved tube of another form. This statement is borne out by recent tests comparing the performance of various draft tubes.

CONCLUSIONS REGARDING FLOW IN THE DRAFT TUBE

42 When the turbine is operating at normal ϕ, the whirling component of the water coming off the runner is in the direction of rotation of the turbine. In all readings of the deviation of this velocity from the vertical, this direction is taken as positive.

43 From curves of traverses made both with and without the ejector open, it has been found that if the turbine is overgated this whirl may become negative, due to the excess of water passing through the runner. At overload the runner is not moving in the same relation to the water as at normal operation, and the whirl under these conditions may become negative at all but very large values of ϕ. When the turbine is operating at high values of ϕ, the whirl is positive, or nearly so, even though the turbine is overgated.

44 From comparisons of traverses made with and without the ejector open, the following general conclusions may be drawn (see Figs. 14 and 15):

1 The characteristic form of the curve of the angle of deviation from the vertical was not influenced by the water from the ejector

2 The velocity of the water at the outer portion of the draft

tube was increased by the action of the ejector water over values obtained without the ejector open

3 The ejector had little or no effect upon the flow of water coming from the runner up to and including a point about 6½ in. from the center line of the draft tube, or in other words, the inner two-thirds of the draft tube was not affected by the ejector.

4 The action of the runner influenced the direction of the flow of water issuing from the ejector, and not vice versa.

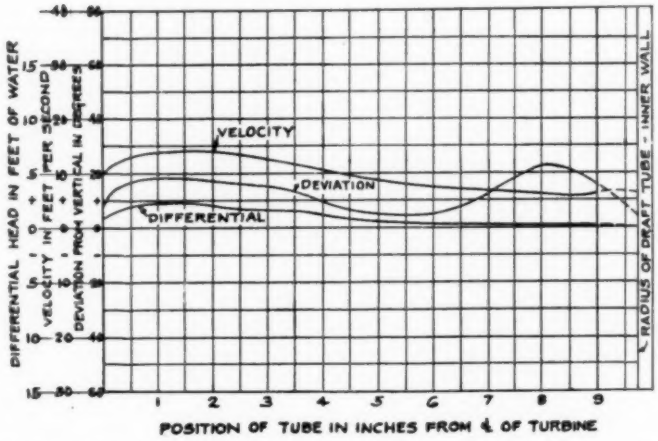


FIG. 14 FLOW IN DRAFT TUBE, EJECTOR CLOSED; GATE OPENING 40 PER CENT; ø = 1.262; RUNNER NO. 65

45 By referring to the curves plotted from results of these traverses, it may be seen that the angle of whirl is very nearly the same regardless of whether the ejecor is open or not, but that the velocity of the water in the outer third of the draft tube is increased considerably over that with the ejector closed.

SUMMARY

46 As was stated in the introduction, the need for the ejector type of turbine is felt only in low-head installations where the reduction in head due to flood periods reduces the capacity of the plant to an appreciable extent.

47 It is obvious that instead of wasting the excess water that

would ordinarily pass over the dam during these periods, the ejector uses this same water to counteract the reduction of the effective head and to maintain the rated output of the plant.

48 While it is imperative that the turbine operate at maximum efficiency when the head is a maximum and water is scarce, it is also true that economical operation is secondary to the maintenance of rated output when there is so great an overabundance of water that the capacity is reduced.

49 The Moody ejector turbine meets these requirements, for

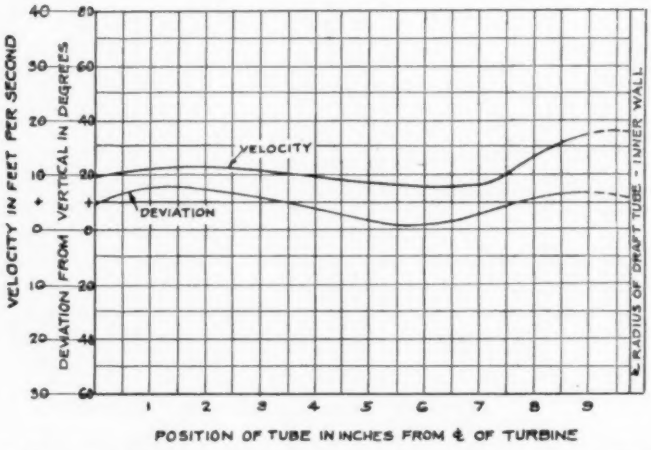


FIG. 15 FLOW IN DRAFT TUBE, EJECTOR OPEN 1.54 IN; GATE OPENING 40 PER CENT; ø = 1.306; RUNNER NO. 65

under design conditions it operates as a normal turbine delivering rated horsepower at maximum efficiency; but when the head is reduced at flood periods, the cylinder gate on the ejector is opened and the output is maintained.

50 This turbine also finds its application, where stream flow and storage conditions permit, in carrying peak loads. At this time the ejector would have the effect as illustrated in the curves of horsepower and efficiency under constant head (Figs. 6 and 9). The maximum possible horsepower that can be obtained is that indicated by the envelope of the several ejector curves. The turbines would therefore be designed to deliver rated power for the average load and rely upon the ejector to carry the peaks. The application

of the ejector to these conditions could only be effected where the peak loads could be accurately estimated in advance.

51 In conclusion, the Moody ejector turbine finds its principal application in low-head installations where the effective head is reduced by flood conditions and great fluctuations occur in the stream flow between high and low water.

No. 1829

A DISCUSSION OF DRAFT TUBE DESIGNS[1]

By Webster K. Ramsey, Aberdeen, Md.
Junior-Member of the Society

THE purpose of this paper is not to present the hydraulic draft tube from a highly theoretical standpoint, but rather from an angle which will be of interest to the greatest number at the present time. It will be surprising to anyone who has studied the subject at all to find what a large number of operating hydraulic engineers have but a limited knowledge of either the importance of the draft tube or the action of the water in the different types.

2 The primary requisite for an understanding of the draft-tube effect upon the water is a knowledge of the action of the water as it issues from the runner of the turbine. In the explanation which follows, a vertical turbine is considered.

[1] The present paper includes a part of a thesis submitted by Messrs. F. M. Rowell, R. A. Snow, M. H. Winchester and the author to the Massachusetts Institute of Technology in May 1921, for the degree of Bachelor of Science, Parts I and II of which respectively dealt with an efficiency test of an Allis-Chalmers vertical hydraulic turbine, and a test of a White hydraucone draft tube. The apparent action of the water in the White regainer as tested led the author to investigate the action of the water in the light of the known conditions existing in other tubes.

It is to be noted that the thesis as submitted, which was practically the same as this paper with the exception of the more detailed description of the Moody spreading tube, was given to the Institute previous to the presentation of the paper on the White hydraucone regainer at the A.S.M.E. Spring Meeting in Chicago. The essence of the discussion following the presentation of the White paper as found in the July 1921 issue of Mechanical Engineering was contained in the above thesis as submitted in the middle of May 1921. See also Paper No. 1794.

The author wishes to acknowledge suggestions from Mr. W. M. White, received in an interview, regarding the construction of a pitot tube and method of testing the hydraucone regainer; and from Mr. L. F. Moody concerning the Moody spreading draft tube.

Presented at the Annual Meeting, December 1921, of The American Society of Mechanical Engineers.

CONDITION OF WATER DISCHARGED FROM TURBINE RUNNER

3 The condition of the water discharged from a runner depends upon the specific speed of the runner. With specific speeds up to 50, which designates a medium-head installation, the water is discharged in such a manner as to cause it to whirl in the same direction as the rotation of the runner, up to best gate. At best gate the water flows practically radially inward. Above best gate the whirl is in the opposite direction to the rotation of the runner. Considering values of specific speed above 50, which designates low-head installations, the velocity of whirl increases in magnitude as the specific speed increases. In this case the water does not change its direction of whirl even after best gate conditions are reached. It is particularly the low-head, high-specific-speed runners having high velocities of whirl which concern the draft-tube requirements, as will be shown later.

4 The following discussion, in order to cover every condition of whirl, considers a runner which

1 Below best gate gives a velocity of whirl in the same direction as the rotation of the runner; decreasing in magnitude as the gate opens
2 At best gate gives practically radially inward flow with no whirl; and
3 Above best gate gives a velocity of whirl in the opposite direction to the rotation of the runner, increasing in magnitude as the gate opens.

THE DRAFT TUBE

5 Turning now to the draft tube, it will be found that this device is used to permit the turbine being placed high enough above tail-water level to give accessibility. In providing an air-tight tube to conduct the water from the turbine runner to the tail race, it is hoped that a good part of the head lost because of the position of the turbine will be regained by the vacuum caused by the weight of the solid column of water flowing downward to the tali race. The problem, however, is not so simple. The water drops with a certain initial axial velocity even at best gate; off best gate there is added a velocity component of whirl. Water is discharged from a runner with a very high exit velocity, especially in the case of a

low-head, high-specific-speed installation. This velocity represents a certain amount of static head acting on the wheel. If this discharged water is allowed to reach the tail race without giving up most of its velocity, then just so much energy will be lost.

6 The purpose of the draft tube, then, is to regain the axial component of velocity and the whirl component as well. In any case the regain, which is usually provided for by supplying a tube of increasing cross-sectional area which regains static head (H_s) from velocity head (H_v) in accordance with Bernoulli's theorem, will be practically impossible unless the water can be made to flow in parallel stream lines and without cross-currents or eddies. The nature of the installation in low-head plants has added one more requirement which demands that the discharged water be turned through an angle of 90 degrees in the shortest possible distance measured in the direction of flow. This latter requirement saves the costly excavation otherwise necessary.

THE PLAIN, SHORT, STRAIGHT DRAFT TUBE

7 Consider a short, straight draft tube discharging into a tail race as shown in Fig. 1. When the turbine is operating below best gate, the water is discharged down the draft tube at an angle. Starting at 0.1 gate, and measuring the angle of flow with respect to the vertical axis of the tube, it is found that the angle of whirl near the sides of the tube is greatest at the lower gates, gradually decreasing until vertical (zero angle) at best gate. If the turbine mechanism is so arranged as to allow the gates to be opened much beyond best gate, the angle will be on the other side of the vertical, showing that the angle of whirl has been reversed, and instead of the water whirling in the same direction as the runner, it will whirl down in the opposite direction.

8 Now, when the water is whirling down a draft tube with great centrifugal force, the action causes water to be drawn up in the central part of the tube. See Fig. 1. This column ascends to a point directly beneath the runner, where it turns over and down and is discharged down the tube again with the water which is being discharged from the turbine.

9 The presence of this column of upward-flowing water is very undesirable in that it increases friction losses as well as reduces the static head or vacuum which would be formed and maintained directly below the runner.

10 It is important to note at this point that when the runner is discharging water at best gate, the water is flowing down in practically straight stream lines. Under this condition, there is no centrifugal action to cause the water to be drawn up from the tail race. There is, however, a loss due to sudden expansion at the discharge end of the tube.

11 It is also to be noted that in a tube of this type there is practically no provision made for straightening out the water flow and thereby utilizing even the short length of tube which has an

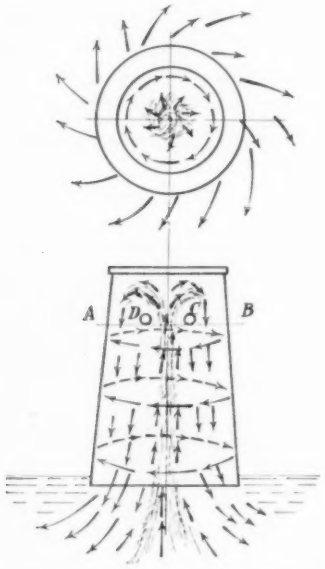


FIG. 1 SHORT, STRAIGHT DRAFT TUBE

increasing cross-sectional area. Further, no provision is made for turning the water through 90 degrees, which is one of the chief requirements of a low-head installation.

FLARING DRAFT TUBES

12 Until the advent of the radial draft tubes the usual method of discharging water from a low-head turbine was to provide a flaring tube having a short 90-deg. bend as shown in Fig. 2.

13 If the turbine was operated always at best gate, it is easy to see that the direction of the water would be changed by friction

along AB. Assuming that a full tube section of water started from C to D, it is to be noted that this water would pass points M and O with the same velocity. Assuming the runner discharges water at an equal rate over the entire area at C, it can be seen that as the water turns the 90-deg. bend the inner particles flowing along OP will not have the same velocity as those flowing along MN. Hence an eddy loss — a loss caused by cross-currents set up by the sliding of one layer of water over another — is present in this form of tube, even

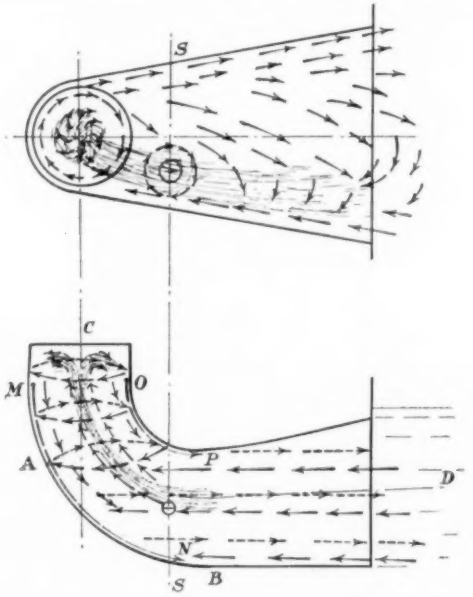


FIG. 2 FLARING DRAFT TUBE WITH SHORT BODY

when operating at best gate. It has been further found that when straight stream-line conditions are obtained, the high-velocity water will flow along the bottom half of the horizontal portion of the tube and discharge below the point D. In not a few cases it has been found that when the upper portion of the flare was a little too high, the water actually flowed back into the tube above the line D.

14 Considering actual operation, a turbine is almost always enough off best gate to cause considerable whirling of the water. This whirling body of water possesses properties not unlike those of the rotating wheel of a gyroscope, and like a gyroscope it tends to keep in a plane perpendicular to its initial axis. Hence it will be

seen that the water will not continue to rotate in a plane perpendicular to the central axis of the draft tube upon reaching the bend in the tube. This theory is substantiated by noting that the drawing action caused by the commotion of the water at the bend, as shown in Fig. 2, renders a part of the discharge area ineffective. In most cases the water actually flows back into the draft tube and is drawn from the points of most violent eddying (marked "+")

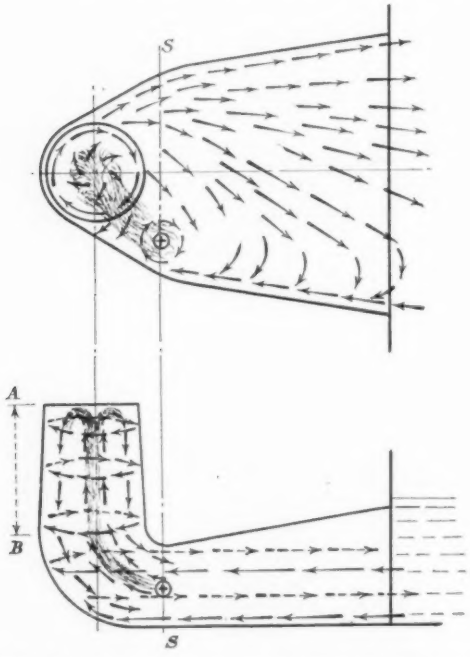


FIG. 3 FLARING DRAFT TUBE WITH LONG BODY

up into the vertical portion as shown. In not a few cases the combined effect of the actions stated causes a complete loss of the total draft head available.

15 The construction of this type is not a factor in its favor. The best of these designs are made in concrete. When made of steel, the tube should bend smoothly at the elbow and not be made up of flat sections of steel plate.

16 Fig. 3 shows a draft tube similar to that of Fig. 2 with the exception of the longer body from *A* to *B*. This longer body has the effect of straightening the flow lines of the water, thus rendering

it in a better condition for regaining through the diverging tube. It is evident that this design requires a greater distance from the horizontal center line of the turbine runner to the bottom of the tail race. The beneficial effect of the added length AB is actually very small.

THE HYDRAUCONE

17 Mr. W. M. White, manager and chief engineer of the hydraulic department of the Allis-Chalmers Manufacturing Company at Milwaukee, Wis., was the first to patent a means whereby static head could be regained from velocity head, and the direction of

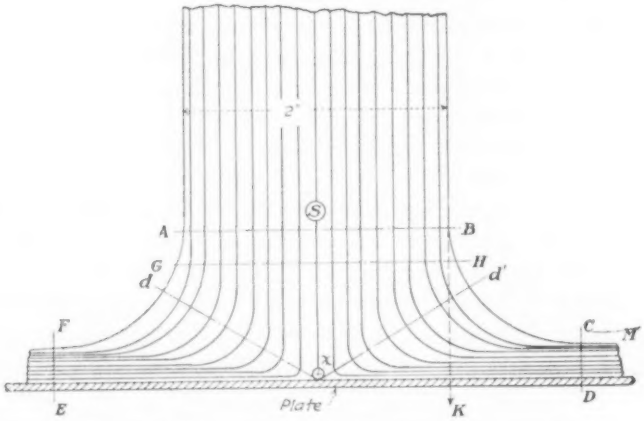


FIG. 4 HYDRAUCONE AND HYDRAUCONE ACTION

water be changed at the same time, in the shortest possible distance measured in the direction of the flow of water. These changes are brought about by what is called by the inventor the "hydraucone action" of the water, together with the regaining qualities of the design which act in accordance with Bernoulli's theorem.

18 The "hydraucone" is that portion of water bounded by the surfaces AB and CD–EF, Fig. 4, and its action, in the case of the straight jet normal to the plate, is to turn the water through ninety degrees without losing any velocity, and at the same time preserving parallel stream lines. The velocity head, moving vertically downward, is converted into static head, and this head used to reconvert the static head into velocity head again, but in a horizontal direction with a very high efficiency.

THE WHITE HYDRAUCONE DRAFT TUBE

19 Realizing the importance of the hydraucone action of the water under the above conditions, Mr. White conceived the idea of utilizing the action to changing the direction of the water flowing from low-head turbines where expense of excavating for a long, straight draft tube would be prohibitive. The development of the White hydraucone draft tube is herein explained.

20 Referring to Fig. 4, a jet of water is shown impinging on a flat plate. The flat plate, which is the base of the hydraucone chamber, is made preferably circular and placed concentric with respect to the discharging jet.

21 Now, if the conoidal surface of the hydraucone, as *ABCF*, were enclosed in a closely fitting chamber, surface friction would result. Hence, a conoidal chamber is so constructed as to be slightly larger than the volume of a free hydraucone. At this point we have an enclosed hydraucone possessing all the characteristics of a free hydraucone. A step further is taken. In order to regain pressure from velocity it is necessary to provide a tube of increasing cross-sectional area. Therefore the conoidal chamber is not only made large enough for a free hydraucone with provision for friction, but also is slightly flared at the lower end, thus combining in one chamber the direction-changing action of the hydraucone with the pressure-regaining action of the section of increasing cross-sectional area.

22 The complete development of the design is shown in Fig. 5, a study of which will show how the pressure is regained in the following four distinct steps:

1 First regain due to the diverging draft tube
2 Second regain due to the diverging hydraucone chamber
3 Third regain due to increasing area as water flows radially outward along the plate
4 Fourth regain due to the diverging draft tube which leads to the tail race
5 Collecting-chamber under the hydraucone baseplate. It serves as a place for collecting all the water coming off the plate and directing this water out through the section (4) into the tail race.

TEST ON THE WHITE HYDRAUCONE DRAFT TUBE

23 The discussion of the hydraucone up to this point has considered that the water issuing from the runner was moving in a radially inward direction and falling thence vertically down the draft tube. In actual practice, as before stated, this is seldom the condition. It is necessary, therefore, that we make a test to ascertain just what happens in a hydraucone-equipped draft tube when the

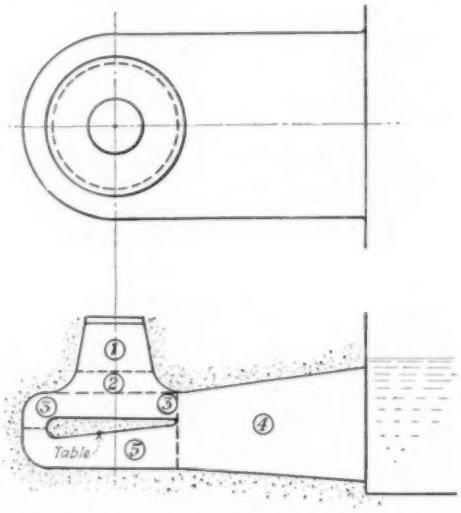


FIG. 5 WHITE HYDRAUCONE REGAINER AS PATENTED

water does not drop vertically downward like a jet, but comes whirling down at a sharp angle.

24 The draft tube on which this test was run was a part of an Allis-Chalmers 1875-kva. hydroelectric unit at Greggs Falls, N. H. The draft tube is equipped with a White hydraucone tube as shown in Fig. 6.

25 Ten tests were made in the following manner: The turbine was brought up to speed (257 r.p.m.), and by means of the load-limiting device on the back of the Allis-Chalmers governor, the gate was allowed to open to 0.1 gate. A special long pitot tube was then pushed into the draft tube until the tip was at the extreme opposite zone of the section. Both static-head and velocity-head hoses were connected to the U-tube of mercury. Knowing that the water was

whirling down the draft tube at an angle, the exact position of the normal element of water was ascertained by turning the nozzle of the tube until the maximum velocity-head reading was obtained. At this position the velocity head and the angle of the nozzle were read. Holding the tube firmly, the velocity hose was disconnected and the static head read for that position. The tip was then drawn back into the next zone, of which there were ten, and readings taken in a similar manner. Having traversed the draft tube across the diameter at 0.1 gate, the turbine was brought up to 0.2 gate (speed

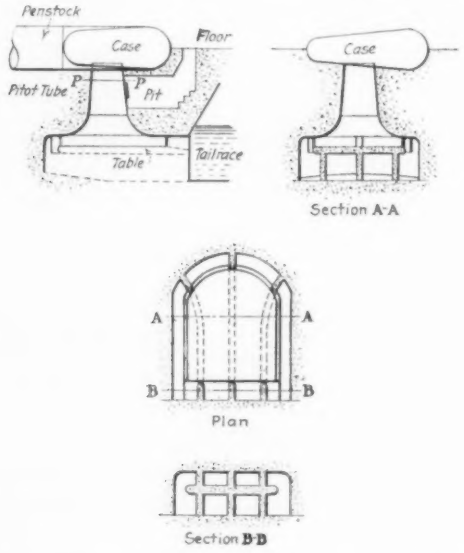


FIG. 6 WHITE HYDRAUCONE REGAINER AS INSTALLED AT GREGGS FALLS, N. H.

constant throughout) and readings obtained in the same way as at 0.1 gate. In like manner readings were recorded for each tenth gate up to maximum gate opening (1.0).

26 An inspection of the curves of Fig. 7 shows that negative velocity-head (H_v) and high static-head (H_s) readings were obtained in the central area of the draft tube at the lower gates, gradually diminishing in magnitude until at 0.9 and 1.0 gates all the velocity-head readings were positive and all the static-head readings were of not unusual magnitude.

27 The results show further that the angle the nozzle made with the vertical axis of the draft tube was greatest at the lower

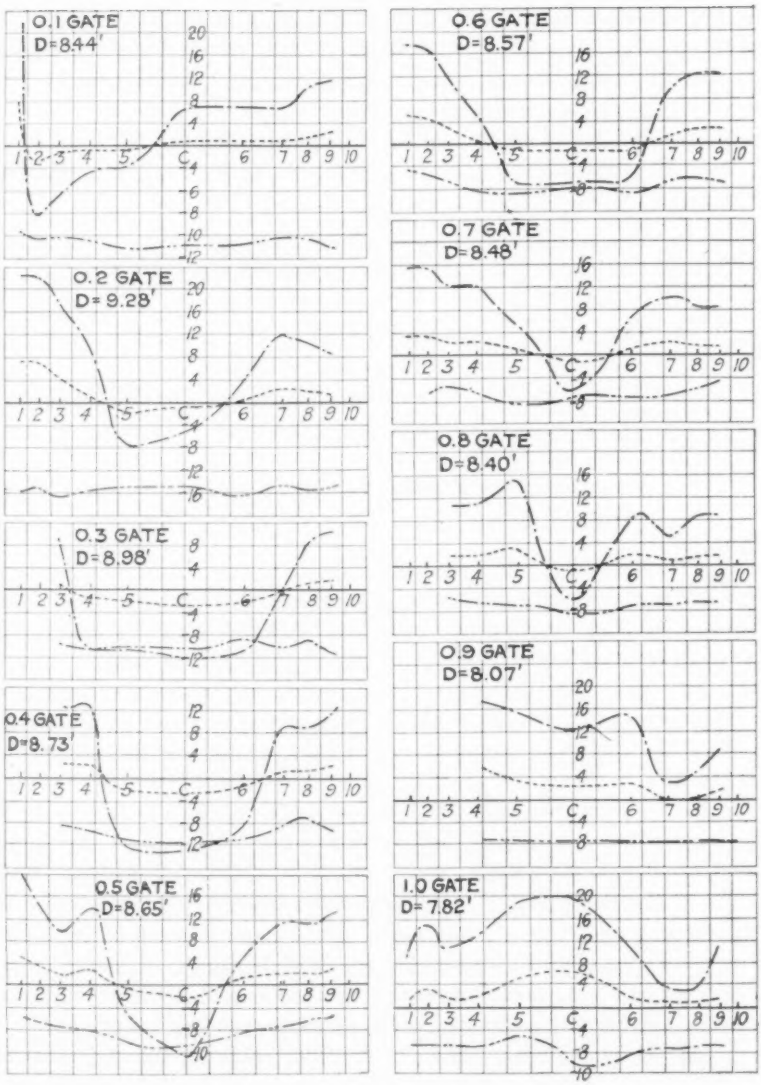


FIG. 7 VARIATION OF STATIC HEAD AND VELOCITY HEAD AS MEASURED ACROSS DRAFT-TUBE DIAMETER FOR DIFFERENT GATE OPENINGS

— — — — — —Velocity of static head in feet of water
— · — · — · —Static head in feet of water
— · · — · · — · · Velocity (ft. per sec.) as calculated from velocity-head curve
D = distance from center line of pitot tube to tail-water level.

gates. It was also noticed that at the gates showing negative H_v readings the angle was greatest at the side of the draft tube and gradually decreased until, at the point of negative readings in the central area, the nozzle was vertical.

28 From the results of the readings taken by the use of a pitot tube traversed across the draft tube as previously explained, the following are noted:

a The velocity and angle of whirl of the water in the draft tube are greatest at low gates, decreasing as the gate is opened

b The negative velocities occur at the low gates when the angle of whirl of the water is greatest

c When negative velocity-head readings are obtained, the pitot-tube nozzle was vertical and in the central area of the draft tube

d The vacuum is greatest at lowest gate, decreasing as the gate is opened.

29 Recalling the conditions which produce the negative velocity-head readings in the case of the short, straight draft tube, similar readings on this test appear to indicate that the column of upward-moving water is still present in this type of draft tube.

30 The construction of the hydraucone-chamber baseplate seems to indicate that it would be impossible for water to be sucked out of the tail race across the top of the plate and thence up the draft tube.

31 A more logical explanation as to the source of the upward-moving column of water is that water is picked up off the inner surface of the discharged water which is whirling down the draft tube as shown in Fig. 8. It is quite conceivable that the water is whirling around the circumference of the inner surface of the draft tube, leaving plenty of space in which the column of water could rise. At any rate there exists this upward-moving column of water in this design when the turbine is operating at the lower gates.

32 It is now necessary to consider whether the hydraucone action of the water is still present. In view of the foregoing paragraphs, the writer is inclined to doubt it when the turbine is operating off best gate and the water is whirling down the tube. The water now impinging on the plate is not a straight jet and therefore the action of a straight jet does not hold. With a straight jet, the force which builds up the static head at x, Fig. 4, is the inertia of the water

falling vertically downward in a straight line all around that point. The force which prevents particle B, for instance, from continuing in a straight line vertically downward to K is the force which reacts at x and helps maintain the high static head at x.

33 Now the whirling water has the opposite tendency to the

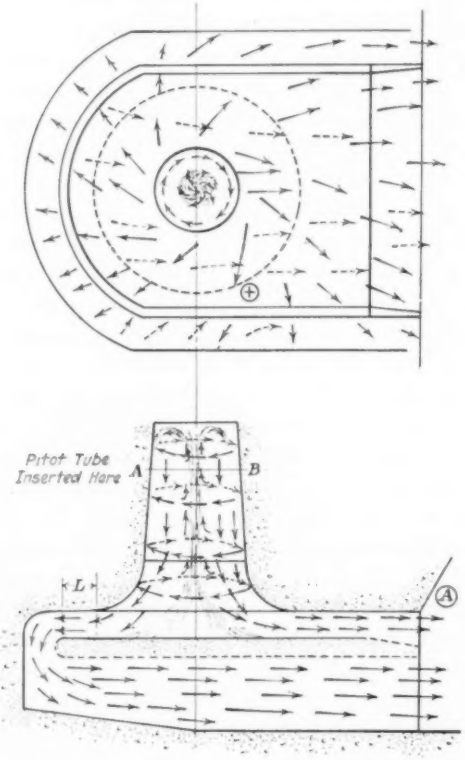


FIG. 8 ACTION OF WATER IN THE GREGGS FALLS TUBE

above, namely, a tendency to fly off at a tangent, out through the annular passage through area CD–EF, Fig. 4; or space (3), Fig. 5, and thus create a tendency to produce rather a vacuum at x than a pressure, or at any other point within the draft tube. From this it is evident that the hydraucone is probably not formed at any but the best gate.

34 Notwithstanding the foregoing reasoning which seems to indicate that it would be impossible to maintain a hydraucone under

whirling conditions of water, the makers state that they have made exhaustive investigations regarding this phase of the subject and have found that the hydraucone action is present under *any* condition of whirl, but that the pressure at the center of the plate may decrease to some extent, such decrease being dependent upon the velocity of whirl.

If If we assume that the hydraucone action is not present at any but the best gate, it is necessary to explain just what are the advantages of this design. As far as the hydraucone is concerned, the writer is inclined to believe that there is very little advantage. There is, however, a gain which more than compensates for the loss of hydraucone effect and which is explained as follows:

36 In order to regain static head from velocity head efficiently, and regain the energy component of whirl, the filaments of water must be made to straighten out and flow in parallel stream lines. The results of the hydraucone draft-tube test show that the lower the gate opening the worse the eddying and the greater the angle of whirl; hence no material effect from either the hydraucone action or the increasing area can be expected from sections (1) and (2), Fig. 5. Yet it will be noted that at 0.1 gate and 0.2 gate the vacuum was very great. This is explained by recalling that water, like any mass, flies off at a tangent as soon as the restraining force causing it to rotate in a circle is released. Thus, when the whirling water reaches the annular opening (3) it rushes out through the passage with great velocity. The water is then flowing in straight stream lines and is in the proper condition for maximum regain of pressure head by means of the increasing area as the water moves radially outward along the plate, and finally through the diverging tube (4) into the tail race.

37 The water passing out through the annular space (3) has considerable energy due to the velocity with which it is thrown out. The shape of the annular passage is of increasing cross-sectional area not unlike that of a water ejector in action, and it is because of the suction caused by this ejector action that such high vacuum readings were obtained. That the foregoing action takes place is substantiated by noting that as the gate is opened the speed and angle of the water discharging down the draft tube decrease, as also does the vacuum, until at best gate (0.9) the vacuum is normal and the centrifugal action is replaced by the hydraucone action.

38 That the resulting loss of the hydraucone action is of less magnitude than the gain due to the ejector effect is proved by tests

which show that the draft-tube efficiencies with this type of hydraucone-chamber construction increase as the gate is closed.

39 While the hydraucone action of the water was the cause for the particular design of Mr. White's, the writer believes that the design has won merit far more because of its ejector action than its hydraucone qualities, in so far as the hydraucone qualities have to do with a regain in the hydraucone chamber, especially under whirling conditions.

40 An interesting coincidence is to be noted with regard to the surging and other noises within the draft tube at different gates. In connection with the pitot-tube traverses, the corresponding surging or rattling within the tube was noted for the different gates:

0.1 gate	Light surging as if made by small amount of water
0.2 gate	About same as 0.1 gate
0.3 gate	Surging 64 times per minute. Runner rattles
0.4 gate	Same as 0.3
0.5 gate	Surging partly ceased. Pitot tube vibrates
0.6 gate	Surging 72 times per min. Not so loud. No rattling of pitot tube
0.7 gate	Steady flow
0.8 gate	Steady flow
0.9 gate	Steady flow
Full gate	Steady flow.

41 It is interesting to note that the upward column of water begins to be very small at 0.7 gate. At this gate the surging ceases and steady flow is maintained.

42 Suggesting a cause for surging at lower gates, it may be said that the tube is not full of water. Then, if the turbine gates close a slight amount suddenly, the vacuum under the runner may lift the volume of water, which partially fills the tube, up toward the runner. The water, falling back again, may readily set up an oscillating movement which will continue until an equal force in the opposite direction happens to occur by the sudden partial closing of the gates as the surface of the oscillating volume of water is on the down "stroke."

43 Whatever the cause, it is certain that such surging is highly undesirable in that it causes an uneven pressure on the runner, thus imposing added duty on the governor and greater strains on the runner and penstock.

WHITE LOW-HEAD HYDRAUCONE REGAINER

44 Fig. 9 shows the White design for low-head installations. The base of the hydraucone chamber in this case is the bottom of the excavation, the collecting chamber being above the table top rather than below. This design may be used for extreme low-head installations. The space marked *s* on this plate corresponds to the space (3) in Fig. 5.

45 Horizontal turbines have been usually arranged to dis-

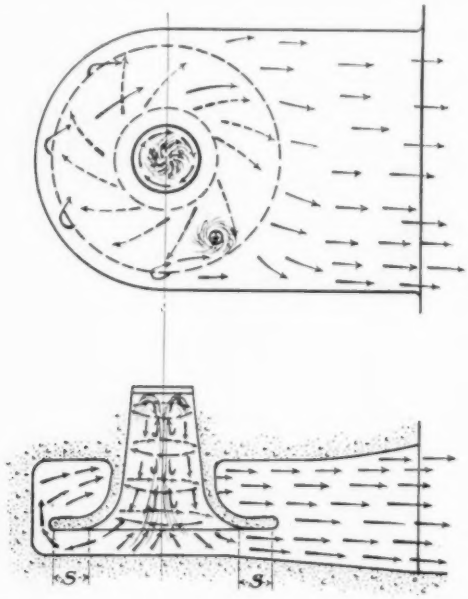


FIG. 9 WHITE LOW-HEAD HYDRAUCONE REGAINER

charge the water into a sharp 90-deg. elbow placed at the top of a vertical draft tube. From what has been already said concerning the gyroscopic action of the whirling water, it may be readily seen that very poor draft-tube efficiencies are to be had with a sharp bend so close to the runner. The advantages of the hydraucone chamber draft tube on such horizontal turbines are obvious.

THE MOODY SPREADING DRAFT TUBE

46 Mr. L. F. Moody, consulting engineer of the I. P. Morris department of the Wm. Cramp & Sons Ship and Engine Building

Company at Philadelphia, has designed a draft tube which, at first glance, bears a striking resemblance to the low-head design by Mr. White. See Figs. 9 and 10. The two designs appear to be practically identical, with the exception of the concrete cone in the Moody design. There is, however, a radical difference in the principles upon which the tubes are based.

47 The Moody spreading draft tube is based on the principle

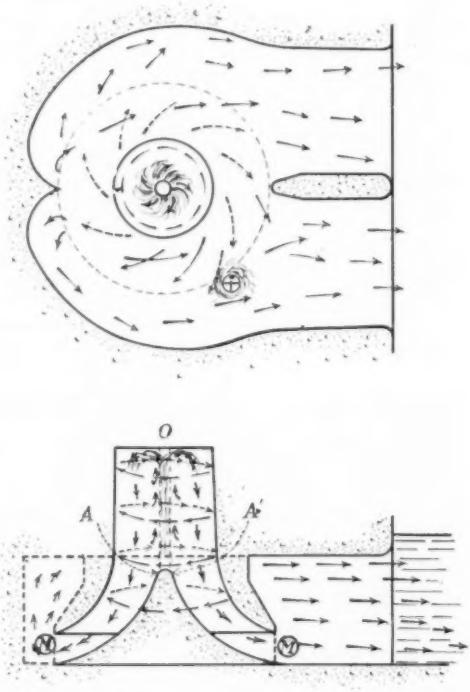


FIG. 10 THE MOODY SPREADING DRAFT TUBE

illustrated by the familiar case of a free vortex in an open basin. Fig. 11 is a diagram of a cross-section through a free vortex. The water, as it whirls through the opening *OS*, assumes a definite shape as shown by the solid lines *ABCDEFGH*. Mr. Moody has shown, with reference to the free flow of water through a space of revolution, that the velocity of whirl varies inversely as the radius and that the corresponding velocity head varies inversely as the square of the radius. If it is assumed, for instance, that the particle x

is on the surface at A, it will increase its velocity of whirl as it approaches CD where the radius is smallest. The smaller the radius, the greater the velocity of whirl.

48 Since the reverse conditions hold — the greater the radius, the less the velocity of whirl — it occurred to Mr. Moody that if the water could be conducted even but a short distance away from the central axis of the draft tube without incurring eddy losses, the

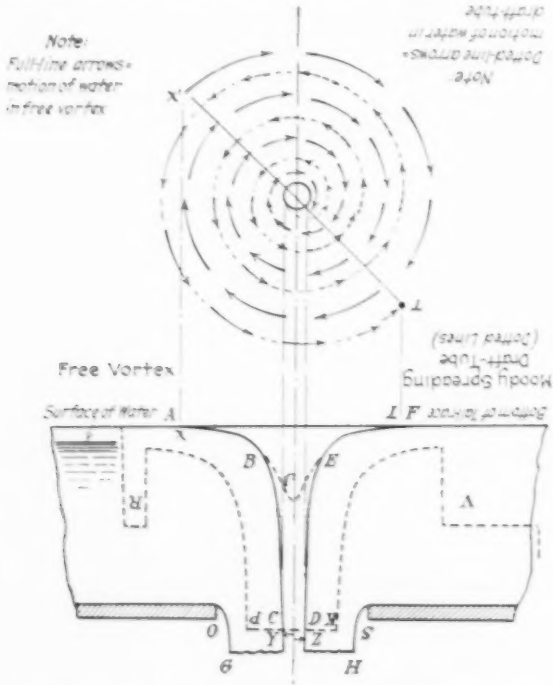


FIG. 11 PRINCIPLE OF FREE VORTEX ON WHICH MOODY TUBE IS BASED

velocity of whirl of the water discharging from the runner could be effectively regained into static head. In other words, if the action of the free vortex could be reversed, the regain could be accomplished.

49 Turn Fig. 11 upside down and consider the conditions now as reversed. The hole OS is discharging water which is whirling at high velocity as shown by the dash and double-dot lines in the lower drawing. Assume that the particle Z is moving toward T. The greater its radius, with respect to the axis, the slower its velocity.

Thus it is evident that the required effect is produced. It now remains to suitably enclose the whirling water and conduct it far enough away from the axis to give the desired reduction in velocity.

50 The dotted lines in this figure represent an enclosed reversed vortex. The outline is that of the Moody spreading draft tube as installed at Niagara Falls. The concrete cone TJX does not fill the entire space which would occur in the case of a free vortex, because such a structure in a tube of this size would be unstable. However, it is recommended that the cone be carried up to the runner in all cases where practicable. The space RV is the collecting chamber into which the draft tube discharges.

51 Now if it is considered that a turbine runner is discharging water with a high velocity of whirl at KP, the water will flow spirally around the circumference of the tube, steadily decreasing its velocity as it increases its distance radially from the axis in accordance with the law previously explained. Thus the Moody spreading draft tube, based on the principle of the free vortex, regains static head from velocity of whirl as well as from velocity of discharge.

CONCLUSIONS

52 The extremely low-head installations which are now being considered have made it necessary to investigate thoroughly a method of regaining all the draft head possible. Since a low-head plant requires a runner of high specific speed, and therefore a high exit velocity from the runner, it is evident that the question of converting velocity head of the water revolving at high speed into static head is a question of great importance. Generally, when low-head plants are built, the expense of excavating for a long draft tube would be prohibitive; hence some form of radial draft tube is preferable.

53 The tests on the radial tubes as against the older designs prove conclusively that the radial type is a long step forward.

54 The writer suggests that an investigation as to the operation of both designs under service conditions be made before attempting a conclusion as to which of the two radial tubes is the better. It is further suggested that the investigation include some data as to the cost of construction of each form.

DISCUSSION

FORREST NAGLER. The paper has been read by the writer with the greatest of interest as was the original thesis of which the present paper forms a part. The entire work was carried out with a remarkable degree of thoroughness. The original thesis was particularly interesting as it is one of the few examples of research work of this kind applied to commercial developments but untainted in any way by commercial consideration. The authors are to be complimented on the extent to which they applied and carried out original lines of thought in investigating the commercial conditions they found.

The writer feels that the portion isolated in the discussion on draft-tube designs is a little bit misleading, in that most of it shows, and the greater portion of the discussion deals with abnormally high or low gate openings as contrasted to the normal gate opening for which the whirl is the least. This may lead to the conclusion that the basic consideration in the design of turbine settings is that of whirling water. This plays an important part but it should not be allowed to completely dominate the problem. This feature of whirl originally gave rise to the concentric form of draft tube but the primary consideration is to secure the maximum efficiency possible at the best gate, and the secondary consideration is to increase, as much as possible, the efficiency at all gates. Even with the highest specific speed runners whirl is not necessary at the point of best efficiency — although in most forms of runners it is usually present.

The writer is inclined to disagree with Mr. Ramsey's discussion that "when water is whirling down a draft tube with great centrifugal force the action causes water to be drawn up in the central part of the tube." It is possible that the tube may be filled by such an action but the rapid circulation so often found in the central portion of the column is motion probably attributable to the pumping action set up in the central portion of the runner, which under small gates is more or less dead so far as developing power is concerned. All runners having radial passages are apt to show this pumping characteristic at small gates. As the blades of the runner are made more nearly radial or propeller like, it becomes easier to eliminate this pumping action. Conversely this pumping action exists to a greater or less extent even at best gate on most runners of the mixed flow type.

The assumption which Mr. Ramsey makes as to the absence

of hydraucone action when there is any whirl in the water is hardly according to facts. Water that is in a state of rotation does have tendency to fly off in a tangent as is stated but this tendency does not have a very great effect on the form of the enclosing chamber for the degrees of whirl usually met with in turbines. In other words, the form of a tube which will just enclose the form which a free whirling jet of water will take is so little different from the form of a natural straight flow hydraucone chamber that the efficiency of the latter is affected but slightly due to the angle of whirl from the runner. This is due to the fact that the natural outline of water, whirling even so much as 30 deg., more nearly approximates the form of the axial portion of the hydraucone than it does the radial portion. The conclusions in the paper were evidently arrived at from lack of realization that the motion in the tube is more forward than rotational and it is the forward component that directs the water in a cylindrical or conical shape as contrasted to a minor whirl component which of itself would cause the water to be thrown off radially.

The secondary ejector action from the radial portion of the chamber advanced by Mr. Ramsey as the explanation of the efficiency of the tube in question has formed a part of all the hydraucone designs. The original experiments were started largely for the purpose of utilizing whirling components more fully and especially to eliminate the handicap that whirl imposed on the curved type of tube. Some of the original hydraucones were made with the idea that practically the entire diffusion was to be obtained in an annular passage in which the water flowed radially outward.

It was naturally impossible to explain to Mr. Ramsey, prior to and during the progress of the test in question, all of the theories, studies and experiments that led up to the perfection of the hydraucone regainer. Apparently this feature of taking advantage of the radial outward flow was not fully covered in the discussions which were held.

Mr. Ramsey's paper falls into a common error of limiting the application of the hydraucone to flow against a flat plate. The paper [1] dealing with the hydraucone, shows many settings — experimental and otherwise, with the impinging surfaces varying widely from a flat surface. These vary from conical to concave. Cones were found to be advantageous in certain instances, but for medium specific speed runners were found to have no appreciable

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Page 1239, footnote, read "See paper No. 1794" instead of "No. 1791."

advantage. Some runners of high specific speeds operate more advantageously with a conical center, and for that reason have been used with such installations. Hydraucones with other than flat surfaces are described quite fully in the original definition of the hydraucone action of water which Mr. White states in his patent to mean "that action of water which occurs as the stream impinges against a surface and is deflected therealong," also — "in the drawings I have shown a hydraucone chamber with a flat bottom for impinging surface but such impinging surface may be conical, convex or concave but is preferably concentric with the axis of the entering stream." The form of the free hydraucone naturally depends not only on the form of the impinging surface but upon the velocity and whirl in the water.

It is interesting to note how closely the Moody Spreading Tube appended to the main portion of Mr. Ramsey's article conforms to the above definitions.

In view of the space devoted to abnormal flow conditions in the Gregg Falls draft tubes investigated by Mr. Ramsey, it may be appropriate to call attention to the fact that the turbines in question showed best efficiencies around 93 per cent which should compare favorably with the performance of any waterwheel ever constructed. This is particularly true when it is considered that the turbines are comparatively small in capacity and dimensions. The writer does not have the exact efficiency curve available and is quoting the above figure from memory. Mr. Ramsey's comments on this feature should be of considerable interest in order to avoid the inference that the hydraulic results obtained at this plant were abnormal or unsatisfactory.

The free vortex theory is interesting and serves as an excellent vehicle on which the spreading tube rides, but to borrow an expression from the legal fraternity the writer would characterize it as "irrelevant, etc." The free vortex shown in Fig. 11 may readily be formed regardless of the depth of the water. The converse application of the principle is not possible however, Fig. 11 itself illustrating this fact in that the depth of water in the free vortex chamber is in decided contrast to the corresponding vertical height of the throat of the spreading draft tube. Likening the diffusion action in the spreading tube to the flow in a vortex where the water is accelerated from a comparative quiet to that existing in a whirling jet is in direct opposition to the well known fact that it is easy to speed up water efficiently in almost any form of passage, but it is

an entirely different problem to diffuse water velocities without excessive losses.

The measure of the success of any of the concentric forms of tubes is found in the closeness with which its outline conforms to that of the true hydraucone. They possess merit primarily on this account and not by reason of the spreading feature, not by reason of the cone in the center — nor by reason of being of the concentric form, although these features may affect the efficiency. More concretely the vertical height of the radial outlet passage near the letter *M* in Fig. 10 must be small in proportion to the diameter, and this is the inherent character of the hydraucone as may be checked by observing the thinness of the sheet of water formed when any jet strikes a smooth surface. If the throat height *M* is varied considerably from the vertical height of the natural hydraucone the efficiency suffers immediately regardless of other characters of the tube. To the writer's knowledge no design has been made whereby commercially good results have been obtained with this type of tube if the throat height is sufficiently great to avoid a direct copying of the hydraucone principle.

THE AUTHOR. The comments by Mr. Nagler are particularly welcomed at this time in that they not only serve to prevent any extreme views concerning the designs of radial tubes but also give rise to other points which were not specifically taken up in the paper.

Mr. Nagler feels that the paper leaves the impression that the basic consideration in the design of turbine settings is that of whirling water. While the author realizes that this should not completely dominate the problem, the fact that the number of high-specific-speed runners having but little whirl is comparatively small, and that the range of gate adjustment under which non-whirling discharge even in these designs is not very great, it is maintained that the consideration of whirl is quite important. While the direct evidence in the form of reference is not at hand at the present writing, the author recalls several instances of draft-tube discussions in engineering journals stating that as much as twenty per cent of the power may be lost by not regaining the energy from velocity of whirl, especially in runners having a specific speed above 70.

With regard to the column of water ascending the central portion of the tube, the author wishes to acknowledge this point of omission in the paper. The circulation of the water, at least near the top of the tube and especially at low gates is largely due to the

pumping action of the runner; the water leaving the lower portions of the runner-blades at high velocities causes a higher vacuum at the top of the runner. Thus the water actually rises to the top of the blades and passes again through the runner. Just how the effect of the pumping action and the centrifugal action divide themselves in producing the column of water at increasing gate opening is a matter of conjecture.

The point taken with respect to assuming that the only tube considered by Mr. White was one with the flat plate is not correct although evident from the omission of consideration in the paper. The author had read Mr. White's June 1921 paper containing the results of tests of conical surfaces, equal-area cones, as well as tests on elbow tubes. The reason for omitting these designs was that it was believed that Mr. White's conclusions had led him to an almost universal adoption of the flat plate as evidenced by the many installations of the flat plate type.

At this point, the author wishes to clear up a point concerning the Allis-Chalmers installation at the Greggs Falls hydroelectric station. In a discussion of such a length as the foregoing paper centered on one particular installation, one is likely to form the opinion that the unit was entirely unsatisfactory. This is absolutely contrary to the facts in this case. As Mr. Nagler mentions, from the curves obtained on the test of the complete Allis-Chalmers unit, including generator, the maximum efficiency of slightly greater than 93 per cent is very near 0.9 gate, and the curve from 0.7 to full gate is very flat showing very good design for a turbine of this size. Regarding operation, the Traction Company reports absolute satisfaction on the two complete units installed. Due to the governing qualities of these units, the usual practice of the company is to "hang" the other stations on the line and make the Allis-Chalmers units do the governing for the entire system.

Returning to the question of radial designs, the White and the Moody are the two most prominent in the field as has already been stated. An obvious point of both designs is that neither the cone nor the flat plate prevents the column of water in the central area of the tube. It is true that the greatest magnitude of water moving upward in the tube is at lower gates and not at best gate. But at such stations as Greggs Falls carrying widely varying loads such as governing, or traction loads, etc., the turbines operate for some considerable periods at even below 0.5 gate.

Now one point yet to be covered is the question of operation.

As already brought out in the main body of the paper, the question of surging in the draft-tube is one of importance especially in large installations. In the Greggs Falls tube, this surging was very pronounced especially when the upward water column was large at the lower gates. Notwithstanding the periodic and more or less violent "jumping" or surging, the action of the governor did not show any signs of increased regulating duty.

In the case of the Niagara Falls tests and installations, it was noted that as regards the efficiencies on the model tubes of the White and Moody designs, there was little choice between the two under straight streamline flow conditions. As for actual operation, it was found that violent surging was present in the White tube at the lower gates, much the same as found at Greggs Falls, to such an extent that an adjustment of tailwater, or an attempt at bleeding air into the top of the tube was considered. The final solution, according to the author's information, was to adjust the tailwater conditions.

In considering both tubes in view of the nearly identical efficiencies under conditions where such efficiencies could be obtained, the choice between the two might well center itself around the question of which of the two has the better operation under actual service conditions.

No. 1830

FLOW IN CONICAL DRAFT TUBES OF VARYING ANGLES

By George E. Lyon,[1] Troy, N.Y.
Non-Member

THE purpose of this investigation was primarily to determine the velocity curves at several cross-sections of straight conical draft tubes. This is of importance as the results will show the flow and its variation along the tube. Incidentally the efficiencies of all but one of the tubes were found.

2 The function of the draft tube is twofold: first, to produce a suction under the turbine equal to the elevation of the turbine above tail water; second, to regain the kinetic energy of the water as it leaves the turbine. A plain cylindrical tube would meet the first requirement provided its mouth was submerged to exclude the air. It is of far greater importance, however, that the second requirement be fulfilled. The high-speed, high-capacity hydraulic turbine of today, operating under low head, discharges water with as much as 25 to 30 per cent of the initial energy of the water in kinetic energy at discharge. Hence it may be seen that without an efficient draft tube the development of much of the potential water power under low head would be seriously handicapped. It is in the design of draft tubes operating under conditions mentioned above that the importance of this investigation may be realized; for, having found the character of flow, a tube may be designed to be more efficient in regaining kinetic energy. Heretofore nothing has been done in the way of finding the character of flow in diverging tubes.

3 For this thesis experiments were performed on seven draft tubes. All of the tubes were 3 in. in diameter at the throat, five enlarging to 6 in. at the mouth and two to 9 in.

[1] Instructor, Dept. of Mechanics, Rensselaer Poly. Inst.

Abstract of a thesis submitted to the trustees and faculty of Rensselaer Polytechnic Institute in conformity with the requirements for the degree of Doctor of Engineering, June 1921, and presented at the Annual Meeting, December 1921, of The American Society of Mechanical Engineers.

4 One tube was of concrete, the others being made of heavy galvanized iron. The angle between opposite elements varied from 4 to 12 deg. Velocity traverses were made in each tube at four cross-sections, including the throat and mouth and at three rates of discharge. The efficiencies of all but one of the tubes were found.

5 This investigation was conducted in the hydraulic laboratory of the Russell Sage Laboratory, Rensselaer Polytechnic Institute, Troy, N.Y., during the school year ending June 1921.

6 Water for the tests was pumped from a large cistern under the hydraulic laboratory by means of a 6-in. Lawrence double-suction centrifugal pump (1000 gal. per min.) direct-connected to an Allis-Chalmers d.c. motor. The water was pumped into a large tank overhead, from which it discharged through an 18-in tee into a 5-in. pipe and thence into a 3-in. pipe through a 3-in. gate valve.

7 After discharging through the draft tube into the concrete-lined flume 30 in. wide by 18 in. deep and containing baffles for smoothing the flow, the water passed over a suppressed weir into the cistern below.

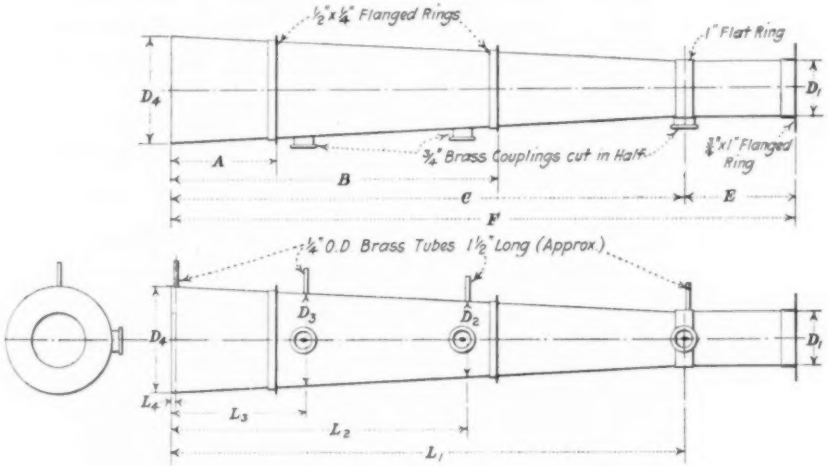
8 The first draft tube (A) was made of concrete, about one part cement to three parts sand. It was formed by pouring the concrete around a wood core in an 8-in. pipe 4 ft. long. The wood core was represented by a cylinder 3 in. in diameter by 5 in. long on the top of a truncated cone having a diameter of 3 in. at the top and 6 in. at the bottom in a length of 43 in. This gave an angle of 4 deg. between sides. This form of tube was very inconvenient, causing trouble in removing the core, also it was very heavy. The rest of the tubes, six in number, were made of galvanized iron and of the dimensions given in Fig. 1. Brass couplings ($\frac{3}{4}$-in.) were soldered to the draft tubes over openings large enough to pass the pitot tube through. These were spaced at equal intervals of cross-sectional area of the draft tubes. The upper section being at the throat and the lower one $\frac{1}{4}$ in. above the mouth. At these four sections piezometer connections were made of $\frac{1}{4}$-in. brass tubes soldered to the draft tubes and at 90 deg. from the pitot-tube connections. These tubes were connected so that the finish was perfectly flush with the inner surface of the draft tubes.

THE TESTS

9 Tests were made at three rates of discharge; 0.985, 0.745, and 0.476 sec-ft. The high rate for draft tube "E" was taken at 0.947 sec-ft. instead of 0.985 sec-ft. All tests were duplicated.

Readings were taken at eleven (sometimes thirteen) points on each diameter. The points were taken closer together at the sides of the tube, spreading out toward the center. In the traverses taken at the throat sections, the throat pressure and tail-water gages were also read, just before taking the pitot-tube readings.

10 Column 1 of the notes (Table 1) gives the radius from the center of the draft tube to the impact orifice of the pitot tube in



TUBE	ANGLE BETWEEN SIDES	A	B	C	E	F	D_1	D_2	D_3	D_4	L_1	L_2	L_3	L_4
B	6°	6"	18½	28.62	6"	34.62"	2.96"	4.25"	5.20"	5.98"	30.19"	16.77"	7.62"	¼"
C	8°	4½	14½	21.25	13.17	34.62	2.92	4.24	5.22	6.00	21.73	12.56	5.75"	¼
D	10°	3	11¾	17.14	17.48	34.62	2.94	4.24	5.24	5.96	17.33	10.02	4.52	¼
E	12°	2	6½	14.27	20.35	34.62	3.00	4.25	5.22	6.00	14.55	8.36	3.82	¼
F	8°	8	20½	42.90	4.35	47.25	3.00	5.76	7.54	9.04	43.40	23.25	10.45	¼
G	10°	6	16½	34.28	12.97	47.25	2.96	5.74	7.60	9.00	34.75	18.56	8.18	¼

Note: Material No. 26 Gal. Iron

FIG. 1 DIMENSIONS OF DRAFT TUBES TESTED

inches. All readings between the two L's being taken on the left of the center line and those between the two R's on the right. Column 2 gives the piezometer readings in inches. In the next column are the pitot-tube readings in inches. These were omitted when the mercury manometer was used. Next are the differences in inches marked y or h for mercury or water, respectively. The next column gives the velocities in feet per second computed from y or h, and the last column gives the hook-gage readings in feet. The zero of the hook gage was 0.766 ft. and the readings for the rates of discharge

TABLE 1 SAMPLE NOTES

Rad. to pitot	Piezometer	Pitot tube	Diff. $=h$	Vel.	Hook gage	Remarks
2.90L	46.50	48.90	2.40	3.58	1.003	Test B-12
2.75	46.10	49.50	3.40	4.25	1.003	
2.50	45.75	50.10	4.35	4.78	1.004	5.98 in. Section
1.75	44.80	51.90	7.10	6.15	1.003	
0.75L	44.20	52.90	8.70	6.76	1.004	Cent pump used
0	43.95	53.35	9.40	7.07	1.003	
0.75R	44.15	53.00	8.85	6.86	1.003	
1.75	44.95	51.60	6.65	5.95	1.003	Water gage used
2.50	46.10	49.60	3.50	4.32	1.003	
2.75	46.45	48.95	2.50	3.65	1.003	
2.90R	46.60	48.75	2.15	3.40	1.002	

Rad. to pitot	Piezometer	D $=y$	Vel.	Hook gage	Flume	Piezometer	Remarks
1.42L	54.20	4.51	17.58	1.002	70.2	6.5	Test D-1
1.32	54.85	5.81	19.95	1.002	70.7	5.9	
1.17	55.20	6.47	21.03	1.002	70.0	6.7	2.94 in. Section
0.92	55.35	6.75	21.50	1.002	68.7	6.2	
0.62L	55.30	6.66	21.35	1.002	68.8	6.1	Cent. pump used
0	55.30	6.66	21.35	1.002	72.0?	1.0?	
0.62R	55.30	6.66	21.35	1.002	66.0	7.2	
0.92	55.25	6.57	21.20	1.002	66.4	6.5	Hg. gage used
1.17	55.10	6.29	20.77	1.002	66.7	6.3	
1.32	54.70	5.51	19.42	1.002	66.5	6.6	
1.42R	54.25	4.61	17.77	1.002	66.7	6.3	
1.42L	54.20	4.51	17.58	1.002	67.2	4.8	Test D-2
1.32	54.75	5.61	19.60	1.002	66.2	7.0	
1.17	55.10	6.29	20.77	1.002	66.8	6.0	2.94 in. Section
0.92	55.25	6.57	21.20	1.002	66.7	6.2	
0.62L	55.25	6.57	21.20	1.002	66.2	7.1	Cent. pump used
0	55.25	6.57	21.20	1.002	66.4	6.9	
0.62R	55.25	6.57	21.20	1.002	66.8	6.2	
0.92	55.25	6.57	21.20	1.002	66.3	7.0	Hg. gage used
1.17	55.10	6.29	20.77	1.002	66.7	6.5	
1.32	54.70	5.51	19.42	1.002	67.1	6.0	
1.42R	54.10	4.31	17.20	1.002	67.1	6.0	

were (see above) 1.112 ft., 0.961 ft., and 0.910 ft., respectively. For the high rate of draft tube "E" the reading was 1.000 ft. Whenever readings were taken at other rates of discharge, the velocity tabulated is that found by correcting the calculated velocity in the ratio of the discharge. The variation in discharge, however, was quite small. The notes taken at the throat sections contain two additional columns. One marked "Flume" gives the reading of the tail

water in inches and the other marked "Piezometer" gives the throat piezometer reading in inches. The average difference of these two columns gives the negative pressure at the throat due to the regain of kinetic energy.

COMPUTATIONS

11 Reduction of mercury readings (y) for velocity:

$v = c\sqrt{2g(13.6 - 1)y/12} = 1.007\sqrt{2 \times 32.2 \times 12.6\, y/12} = 8.276\sqrt{y}$

for pitot tube No. 1.

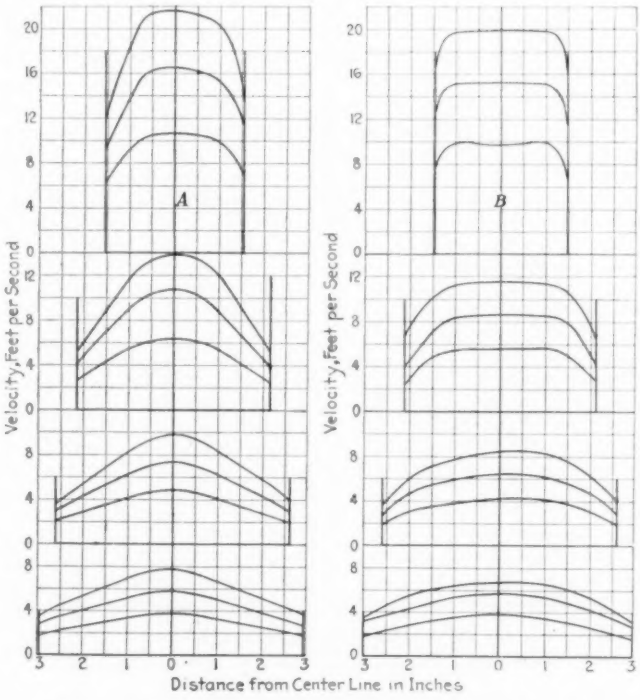


FIG. 2 VELOCITY CURVES OF DRAFT TUBES A AND B

Reduction of water readings (h) for velocity.

$v = c\sqrt{2gh/12} = 1.007\sqrt{2 \times 32.2\, h/12} = 2.331\sqrt{h}$ (tube No. 1)

$v = c\sqrt{2gh/12} = 1.000\sqrt{2 \times 32.2\, h/12} = 2.316\sqrt{h}$ (tube No. 2)

12 The following will serve to indicate in detail the computations, test D-1 (in Table 1) being selected.

13 From the piezometer reading of 54.20 in. on the mercury manometer by previous calibration the difference between the pie-

zometer and pitot-tube readings is calculated to be 4.51 in. The zero reading for these two tubes is 51.90 in., but as the bore was not the same, a slight correction was introduced. $v = 8.276\sqrt{y} = 8.276\sqrt{4.51} = 17.58$ ft. per sec.

14 After plotting the velocity curves against radius, the moment of this area was found to check the discharge and the per cent

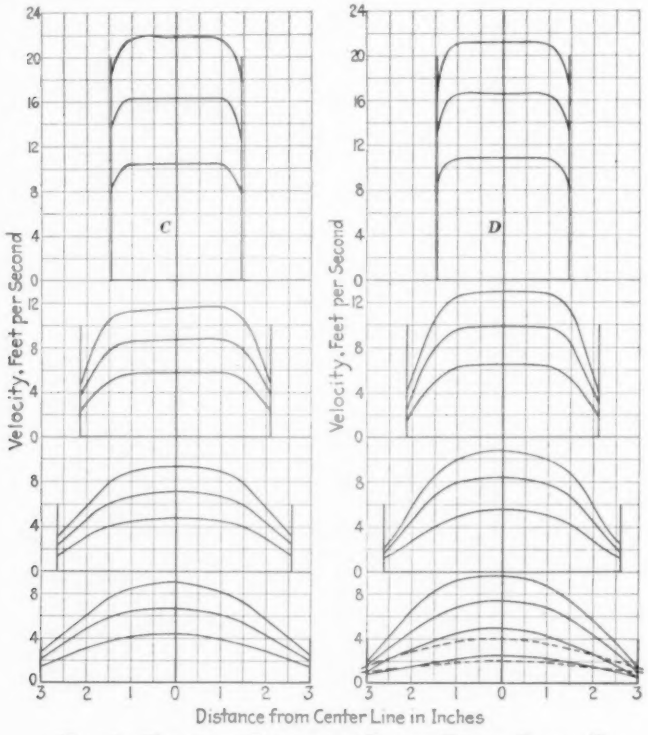


FIG. 3 VELOCITY CURVES OF DRAFT TUBES C AND D

error e tabulated in Tables 2 and 3. As will be explained later, the moment of the v-r curve is equal to $Q/2\pi$. $e = 100\,(Q - Q_0)/Q_0$, where Q is the discharge by integrating velocity curve and Q_0 is the discharge by weir which is taken as being correct. For tests D-1 and D-2, $e = 2.2$ per cent.

15 The average difference of the last two columns of tests D-1 and D-2 gives the negative pressure head in inches at the throat due to regain and is tabulated in Table 2 under "$-p$." In this case $-p = 60.96$ in. From the velocity curve plotted from these tests (Fig. 3)

the kinetic-energy curve was constructed by plotting v^3 against r^2. The area under this curve is the kinetic energy possessed by the water at the throat section. In this case K.E./sec. = 394.0 ft. lb.

The mean velocity head $= \frac{v^2}{2g} = \frac{\text{K.E.}}{\text{sec.}} \frac{1}{wQ_0} \frac{1}{(1+e)^3}$ (see Par. 26)

$$= \frac{394.0}{62.4 \times 0.985} \frac{1}{(1.022)^3} = 6.01 \text{ ft.}$$

The efficiency of regain $= \frac{60.96}{12 \times 6.01} = 84.5$ per cent.

Values of $\frac{\text{K.E.}}{\text{sec.}}$, $\frac{V^2}{2g}$ and efficiency may be found in Table 2.

TABLE 2 THROAT SECTIONS

Draft tube	Q_0 sec-ft.	e per cent	$-p$ in.	K.E.	$V^2/2g$ ft.	Effy. per cent.
A	0.985	−2.7	——	——	——	——
A	0.745	−1.1	——	——	——	——
A	0.476	−1.3	——	——	——	——
B	0.985	−5.3	62.18	331.5	6.35	81.6
B	0.745	−3.0	35.30	146.7	3.46	85.0
B	0.476	−3.8	14.58	39.8	1.51	80.6
C	0.985	−0.6	68.06	424.5	7.04	80.7
C	0.745	+1.2	39.76	178.3	3.71	89.3
C	0.476	−0.2	15.86	46.2	1.567	84.3
D	0.985	+2.2	60.96	394.0	6.01	84.5
D	0.745	+2.0	38.33	188.8	3.68	87.0
D	0.476	+4.0	16.20	52.2	1.566	86.2
E	0.974	+4.5	57.49	391.0	5.69	78.5
E	0.745	+3.9	32.37	173.2	3.33	81.0
E	0.476	+2.1	13.85	46.4	1.475	78.4
F	0.985	−3.6	68.60	344.0	6.25	91.5
F	0.745	−8.9	38.28	123.7	3.52	90.7
F	0.476	−3.4	15.75	38.4	1.437	91.4
G	0.985	−6.7	58.54	316.5	6.35	77.0
G	0.745	−6.0	32.28	133.0	3.44	78.2
G	0.476	−7.1	13.18	33.6	1.41	77.8

16 Consider test B-12 in which the hook gage reading was 1.003 ft. instead of 1.002 ft.

17 The difference $h = 2.40$ in.

$$v = 2.316\sqrt{2.40} \times \frac{0.985}{0.990} = 3.58 \text{ ft. per sec.}$$

0.985 = discharge for hook-gage reading of 1.002 ft. (standard)

0.990 = discharge for hook-gage reading of 1.003 ft.

At this section the diameter on which the readings were taken did

not equal the mean diameter. Hence upon integrating for discharge, the result was modified by multiplying by $\left(\frac{5.98}{6.00}\right)^2$ and the error (e) computed after this correction was made.

18 The various formulas which are mentioned above are derived as follows:

19 The moment of the area of a v-r curve about the center line is equal to the volume of water per second divided by 2π. Let r = radius in feet corresponding to the velocity in feet per second on the

TABLE 3 VALUES OF e FOR SECTIONS BELOW THROAT

Draft tube	1st. Sect. below throat	2nd. Sect. below throat	Sect. at mouth	Q Sec-ft.
A	−7.1	−4.5	+0.5	0.985
A	−3.9	−2.3	+1.9	0.745
A	−2.5	−3.4	+0.8	0.476
B	−1.3	−2.6	+1.7	0.985
B	−4.0	−3.1	+6.6	0.745
B	−2.9	+0.2	+4.4	0.476
C	——	−2.6	+3.7	0.985
C	——	−2.8	+8.7	0.745
C	——	+1.3	+9.4	0.476
D	−1.4	+1.5	+6.6	0.985
D	−2.2	+4.8	+4.8	0.745
D	−1.3	+0.8	+7.7	0.476
E	−0.8	+0.1	+6.7	0.974
E	+0.4	−0.1	+5.4	0.745
E	+1.5	+3.8	+5.7	0.476
F	−2.4	−2.7	+5.6	0.985
F	−2.3	−1.7	+7.2	0.745
F	−2.6	+0.6	——	0.476
G	−3.6	−8.4	+6.8	0.985
G	−1.3	−3.5	+4.0	0.745
G	−1.5	+4.2	——	0.476

v-r curve. Then Q = discharge in cubic feet per second

$$= \int_0^r v 2\pi r dr = 2\pi \int_0^r r v dr.$$

But vdr is d (area) under curve, and $rvdr$ is the moment of this area about the center line. Performing the integration:

$Q = 2\pi \times$ moment of half-area about center line

$= \pi \times$ sum of the moments of the two half-areas about the center line.

20 The efficiency of a draft tube as a regainer of kinetic energy may be determined as follows:

21 Consider a draft tube as shown in Fig. 4, and that the upper tank is raised or lowered until the head h' is sufficient to overcome the losses at a given rate of flow. Thus h' is the total loss. The velocity head in the upper tank is assumed to equal zero and there is no loss down to the throat of the draft tube. Then

$$p_a + z_a = p_b + \frac{V_b^2}{2g} \quad \text{or} \quad \frac{V_b^2}{2g} = z_a - p_b + p_a.$$

Now

$$p_b = -h_3 + p_a \quad \text{and} \quad p_a = 0 \therefore \frac{V^2}{2g} = h_1$$

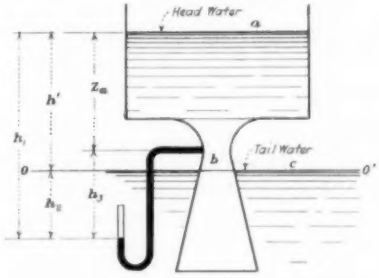


FIG. 4 ARRANGEMENT OF DRAFT TUBE

Now h′ represents the loss, so that the efficiency of a draft tube as a regainer of kinetic energy

$$= \frac{\frac{V_b^2}{2g} - h'}{\frac{V_b^2}{2g}} = \frac{-h_2}{h_1}$$

Readings are referred to atmospheric pressure as a datum. h_2 is negative as the zero of the gage board is at $O - O'$, the surface of tail water, and hence $\frac{-h_2}{V_b^2/2g}$ is positive.

22 Where there is a loss between a and b the efficiency can not be found by simply taking the readings h_1 and h_2. In this case the reading h_1 must be replaced by $V_b^2/2g$ determined from the velocity traverse taken at this section. This was the method used in this investigation.

23 Kinetic energy per second = K.E./sec. = $Wv^2/2g$

$W = \int_0^r wv2\pi r dr$, where w = density of water

$$\frac{\text{K.E.}}{\text{sec.}} = \int_0^r \frac{1}{2}\frac{v^2}{g} 2w\pi\, rvdr = \frac{w\pi}{2g}\int_0^r v^3\, 2rdr = \frac{w\pi}{2g}\int_0^r v^3 d\,(r^2)$$

Hence the area under the v^3r^2-curve multiplied by $\frac{w\pi}{2g}$ equals the kinetic energy per second.

24 To get $V_h^2/2g$, divide by W, since head is the amount of energy per pound of water.

$$\therefore \frac{V_h^2}{2g} = \frac{\text{K.E./sec.}}{W} = \frac{\text{K.E./sec.}}{wQ}$$

Due to the error e found by checking the velocity curves against the weir for discharge, a correction was applied to the velocity head $V_h^2/2g$ just found. This correction is based on the assumption that the velocities found from the pitot-tube traverse must be increased or decreased proportionally so that the discharge found by the "moment" method equals that found by the weir. It will be seen that this is equivalent to changing the constant for the pitot tube.

25 Let Q_0 = discharge by weir (assumed to be correct)

Q = discharge found by taking the moment of the v-r curve

$$Q_0 = kQ$$

$$\text{Error} = e \quad \frac{Q - Q_0}{Q_0} = \frac{1}{k} - 1$$

$$Q = AV \quad Q_0 = AV_0 \quad \therefore \frac{Q}{Q_0} = \frac{V}{V_0} = \frac{1}{k}$$

where v = pitot-tube velocity and v_0 = corrected velocity.

26 Let K.E. = kinetic energy from v^3r^2-curve.

K.E.$_0$ = theoretical kinetic energy from $v^3v_0^2$-curve (if such curve were plotted)

As before $\text{K.E.} = \frac{1}{2}\int_0^r \frac{v^2}{g} 2w\pi rvdr$

$$\text{K.E.}_0 = \frac{1}{2}\int_0^r \frac{v_0^2}{g} 2w\pi rv_0 dr = \frac{1}{2}\int_0^r \frac{k^2v^2}{g} 2w\pi rkvdr$$

$$\therefore \quad \text{K.E.}_0 = k^3\text{K.E.}$$

and

$$\frac{v_0^2}{2g} = \frac{\text{K.E.}_0}{wQ_0} = \frac{k^3\text{K.E.}}{wQ_0} = \frac{\text{K.E.}}{wQ_0}\frac{1}{(1+e)^3}$$

since

$$k = \frac{1}{(1+e)}$$

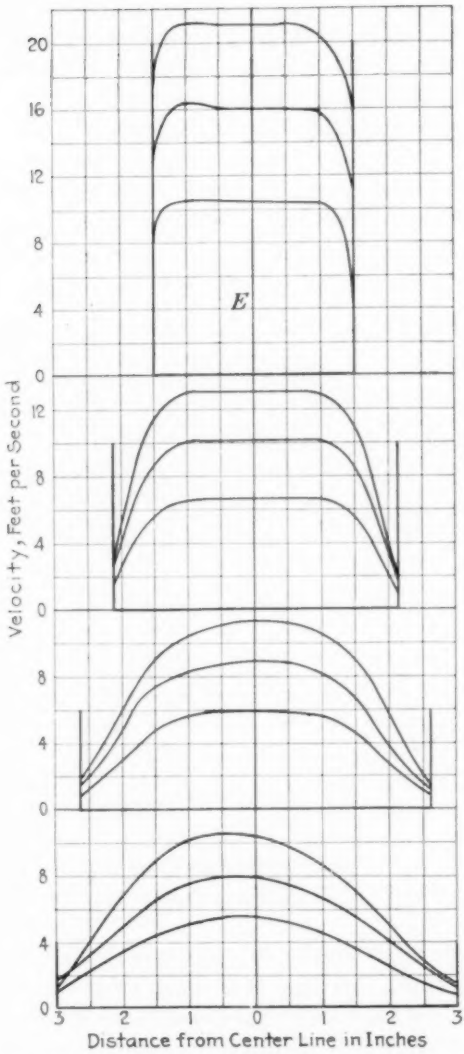


FIG. 5 VELOCITY CURVES OF DRAFT TUBE E

27 Each velocity curve (Figs. 2, 3, 5 and 6) represents the results of two individual traverses or tests. Hence for the 86 curves shown, 172 tests were made. Two additional tests, D-25 and D-26, were made at the mouth of draft tube "D" at a discharge of 0.223

cu. ft. per sec. (hook gage = 1.851 ft.). This is the lowest full-line curve plotted under "D" (Fig. 3). Four more tests, D-27, D-28, D-29 and D-30, were made 6 in. below the mouth of draft tube "D"; the first two at 0.476 sec-ft. and the last two at 0.985 sec-ft. These two curves are shown by the dotted lines (Fig. 3). For the piezom-

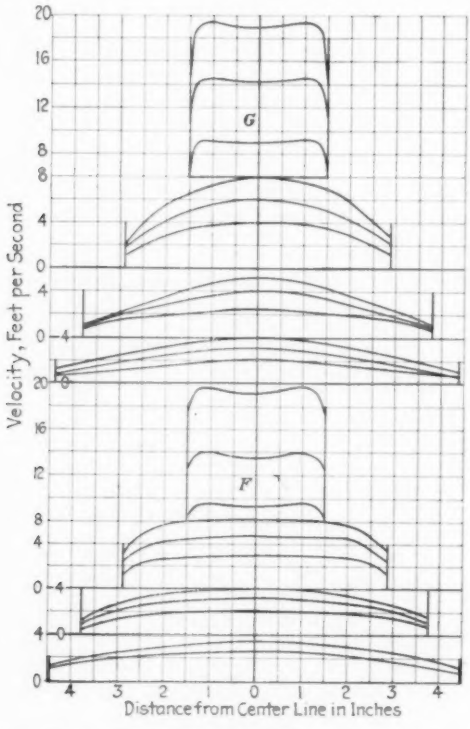


FIG. 6 VELOCITY CURVES OF DRAFT TUBES F AND G

eter reading, the flume connection was used. The efficiency curves (Fig. 7) were plotted on a base of average velocity.

28 The velocity curves for the throat section of draft tube "A" were so distorted it was thought that a sudden enlargement to reduce the velocity would permit the water to be drawn off in a smoother stream-line flow. Accordingly the section of 8-in. pipe with bell mouth at bottom to meet the 3-in section was introduced The effect of this was very noticeable, as the curves for the remaining tubes were very nearly symmetrical.

29 With a couple of exceptions, the curves indicate:

1 That the velocity at the throat (before the water enters the flare) is uniform, falling off at the walls

2 That this uniform velocity breaks down at the sides of the diverging tube first and gradually falls off nearer and nearer the center. The curves for the first section below the throat indicate this

3 That some distance down in the draft tube, the uniform velocity which persisted at the center finally disappears and the velocity curve becomes peaked at the center

4 That this peaked curve persists down the rest of the tube,

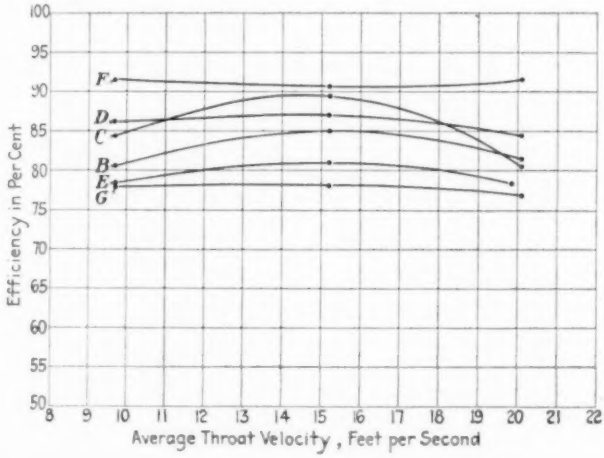


FIG. 7 EFFICIENCY CURVES OF DRAFT TUBES

flattening out somewhat toward the mouth, but even there having a center velocity of from twice to five times the wall velocity

5 That this ratio of center velocity to wall velocity increases with the angle of flare

6 That at a distance below the mouth of the draft tube equal to the diameter at this point, the curve remains peaked. This is only based on the four under-water traverses below draft tube "D."

30 "Lateral flow," the motion of the water due to the flare, appears to be greatest near the wall, breaking down the velocity curve very greatly here. Its effect at the center of the tube is more gradual and, at first, uniform.

31 At the end of the tests the impact orifice in pitot tube No. 1 was plugged and a small hole drilled in the side. A traverse (Table 4) was made, referring the pitot-tube readings to the piezometer at the wall of the tube. It will be seen that the pitot tube records a

TABLE 4 TRAVERSE DRAFT TUBE "D," 4.24 IN. SECTION

Rad. to Pitot	Piezometer	Tube press.	Diff.	Hook gage
2.02L	42.75	42.30	−0.45	0.910
1.92	42.85	42.10	−0.75	0.910
1.72	43.15	41.70	−1.45	0.910
1.42	43.50	41.10	−2.40	0.910
0.82L	43.60	40.90	−2.70	0.910
0	43.65	40.85	−2.80	0.910
0.82R	43.60	41.00	−2.60	0.910
1.42	43.40	41.40	−2.00	0.910
1.72	43.20	41.90	−1.30	0.910
1.92	43.00	42.30	−0.70	0.910
2.02R	42.85	42.60	−0.25	0.910

lower reading than the piezometer. This is not due to whirl as the differences are all negative. The curvature of the pitot tube would not permit its side orifice to record true pressure.

32 It would be of interest in the future to conduct tests under very low rates of discharge, also to determine whether or not there is a variation of pressure across a section.

No. 1831

REPORT OF COMMITTEE ON STANDARD TONNAGE BASIS FOR REFRIGERATION

A SPECIAL committee of the American Society of Mechanical Engineers consisting of Messrs. Fred Matthews, Chairman, Henry Torrance, Jr., Secretary, Charles W. Berry, Peter Neff, John E. Starr, and Gardner T. Voorhees, the membership of which was endorsed by the American Society of Refrigerating Engineers, was recently appointed to fix a "Standard Tonnage Basis for Refrigeration." Its findings reported to The American Society of Mechanical Engineers at its Annual Meeting held on December 7–10 were as follows:

(1) A standard ton of refrigeration is 288,000 B.t.u.

(2) The standard commercial ton of refrigeration is at the rate of 200 B.t.u. per min.

(3) The standard rating of a refrigerating machine[1] using liquefiable vapor is the number of standard commercial tons of refrigeration it performs under adopted refrigerant pressures.[2]

This report has been approved by the American Society of Refrigerating Engineers.

[1] A refrigerating machine is the compressor cylinder of the compression refrigerating system, or the absorber, liquor pump and generator of the absorption refrigerating system.

[2] These pressures are measured outside and within 10 ft. of the refrigerating machine, distances which are measured along the inlet and outlet pipes, respectively; (*a*) the inlet pressure being that which corresponds to a saturation temperature of 5 deg. fahr. (— 15 deg. cent.) and (*b*) the outlet pressure being that which corresponds to a saturation of 86 deg. fahr. (30 deg. cent.)

Presented as a preliminary report at the Annual Meeting, December 1920. Revised report presented and adopted by The American Society of Mechanical Engineers at the Spring Meeting, May 1921.

No. 1832

REPORT OF JOINT COMMITTEE ON STANDARDS FOR ROLLER TRANSMISSION CHAINS

THE Joint Committee on Standards for Roller Transmission Chains of the Society of Automotive Engineers, The American Society of Mechanical Engineers, and the American Gear Manufacturers Association has developed standards for (1) Roller Transmission Chain Dimensions, Heavy, Medium, and Light Series; (2) Roller Transmission Chain Sprockets; (3) Tooth Forms for Roller Chain Sprockets and corresponding cutter design; and (4) Uniform System for Numbering Chains.[1]

2 As its work has progressed, the Joint Committee has kept constantly in touch with the work of the Association of British Driving Chain Manufacturers with a view to the final adoption of standards which would receive international approval. Consequently certain concessions have been made on both sides and the standard dimensions given below are the same, with a very few exceptions, as those adopted by the A.B.D.C.M., and most American chains will operate over British sprockets.

ROLLER TRANSMISSION CHAINS — HEAVY SERIES

3 The "chain widths" given in Column 5 of Table 1 are for the wide-chain series. The numbers express the exact distance between the inside plates and are as near to the norm of $W = \frac{5}{8} P$ as seems advisable in consideration of present practice and the desire to express dimensions in terms of binary fractions. These are known as "wide" chains.

4 The Committee also recognizes as standard a narrow class of chains for which W = approximately 0.41 P, but recommends only the wide series for general practice. These narrow chains can be made in all three series, heavy, medium and light.

[1] Standards (1) and (2) were presnted to and adopted by THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS at the Spring Meeting, May 1921. Standards (3) and (4) were presented to and adopted at the Annual Meeting, December 1921. The S. A. E. has adopted the entire set.

TABLE 1 DIMENSIONS OF ROLLER TRANSMISSION CHAIN, HEAVY SERIES, IN INCHES

Chain No.	Pitch	Roller Diameter	Pin Diameter	Thickness of Inside Plates	Chain Width Min. (W^1)	Measuring Load Lb.	Approx. Test Load Lb.
30 W	$\frac{3}{8}$	0.250	0.125	0.050	0.250	14	464
40 W	$\frac{1}{2}$	0.3125	0.156	0.060	0.3125	25	825
50 W	$\frac{5}{8}$	0.400	0.200	0.080	0.375	39	1280
60 W	$\frac{3}{4}$	0.469	0.234	0.094	0.500	56	1850
80 W	1	0.625	0.3125	0.125	0.625	100	3300
100 W	$1\frac{1}{4}$	0.750	0.375	0.156	0.750	156	5150
120 W	$1\frac{1}{2}$	0.875	0.4375	0.1875	1.000	225	7430
140 W	$1\frac{3}{4}$	1.000	0.500	0.220	1.125	325	10700
160 W	2	1.125	0.5625	0.250	1.250	400	13200
200 W	$2\frac{1}{2}$	1.550	0.781	0.3125	1.5625	625	20600
240 W	3	1.900	0.9375	0.375	1.875	900	29700
320 W	4	2.500	1.250	0.500	2.500	1600	52800
400 W	5	3.000	1.500	0.625	3.000	2500	82500
	For-mulas	Approx. $\frac{1}{4}$ P	Approx. $\frac{1}{3}$ roller diam.	$\frac{1}{8}$ P	Approx. 0.625 P	100 P^2	Approx. 3300 P^2 or $\frac{1}{3}$ breaking strength

[1] See *W* in illustration of Table 2

ROLLER TRANSMISSION CHAINS — MEDIUM SERIES

5 The standard medium-weight series of roller transmission chains is developed by adopting for each pitch the standard rollers, bushings, and pins of the next shorter pitch and assembling them with side plates equal in thickness to those of the next shorter pitch of the heavy series.

ROLLER TRANSMISSION CHAINS — LIGHT SERIES

6 The standard light-weight series of roller transmission chains is developed by adopting for each pitch the standard rollers, bushings, and pins of the second shorter pitch and assembling them with side plates equal in thickness to those of the second shorter pitch of the heavy series.

7 By thus setting the five principal dimensions of the chain for each of the thirteen pitches in each of the three series, the Committee has done all that is necessary to insure the interchangeability of spare links between chains of various makes and also the interchangeability between chains and sprockets.

8 *Tolerance for Chain Length.* New chains under standard measuring load may be one sixty-fourth inch per foot over-length, but must not be under length.

ROLLER TRANSMISSION CHAIN SPROCKETS

9 Table 2 and its accompanying figure give in convenient form the standard dimensions of chain sprockets as far as width, clearance and transverse sections are concerned.

ROLLER TRANSMISSION CHAIN SPROCKET-TOOTH FORM AND SPROCKET-TOOTH CUTTERS

10 *Historical Note.* The first tooth forms for roller-chain sprockets were designed upon the theory that the tooth action was similar to that of gear teeth, the chain being regarded as analogous to a rack. It was, however, soon found that when chains had been in use a short time the wear at the joints caused an elongation of the pitch of the chain, thus destroying perfect registration with the sprocket teeth which resulted in rough action and rapid wear. The sprocket teeth were then redesigned to provide for the chain elongation by making the tooth gap somewhat greater than the diameter of the chain roller, but with little or no modification in the contour of the tooth, which was so shaped that the chain roller bore almost squarely against the sprocket tooth, throwing practically the whole load on a single tooth.

11 Meanwhile, the use of roller chains became more extensive and the demands made on them more exacting. This condition naturally led, both in this country and abroad, to a more careful study of the action between chains and sprocket teeth than had hitherto been given to the subject, and it soon became apparent that by giving some obliquity to the face of the sprocket tooth, quite different and decidedly better results could be obtained. The action of the forces called into play can be readily analyzed by the graphic method of laying out a parallelogram of forces which will show in true proportion the initial tension on the chain, the resultant thrust on the sprocket tooth (determined by the degree of obliquity selected) and a balancing force passed on to the ensuing links of the chain where this action is repeated in a lessening amount.

12 By this method the load on a chain may be distributed over several teeth of the sprocket and at the same time, by extending the oblique portion of the sprocket tooth a suitable distance, it is made possible for a chain of elongated pitch to slide outward on this oblique face until it finds a pitch curve on the sprocket corresponding to the lengthened pitch of the chain. These general findings were freely exchanged among the chain manufacturers

and one of the English companies published a brochure on the subject. During the war more pressing matters prevented an interchange of opinion on this subject, but nevertheless most of the sprocket manufacturers evolved tooth forms along these lines.

SPROCKET TOOTH FORM (SEE FIG. 1)

13 It is now proposed to effect an agreement upon one standard form that shall supersede the various individual designs, and the formula now presented by the Joint Committee for adoption is a composite of the best features of the various tooth shapes that have been tried out and pronounced satisfactory.

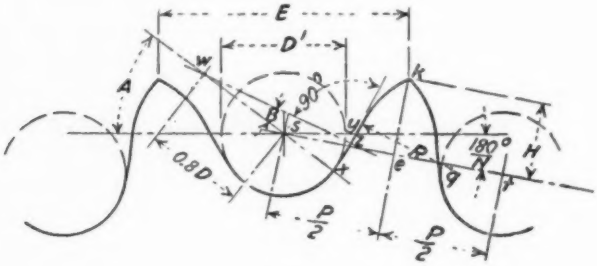


FIG. 1

$$A = 31^\circ + \frac{90^\circ}{N}$$

$$ws = 0.8\,D$$

$$B = 13^\circ - \frac{18^\circ}{N}$$

$$xy = 1.3\,D \times \sin\left(25^\circ - \frac{90^\circ}{N}\right)$$

$$E = O.\,D. \times \sin\frac{180^\circ}{N}$$

$$\text{Outside Diameter of Sprocket} = P\left(0.7 + \cot\frac{180^\circ}{N}\right)$$

P = pitch of chain
D = nominal roller diameter
N = number of teeth in sprocket
D' = diameter of seating curve = $1.005\,D + 0.003$ in.
H = height of tooth crest above the chord st = $0.35\,P$
xz is a circular arc whose center is at w

zy is a straight line perpendicular to wz

ek is a line perpendicular to st

qy is a line parallel to wz with q so located that the circular arc through y will pass through k, thus forming a pointed tooth.

SPROCKET–TOOTH CUTTERS (SEE FIG. 2)

14 Cutters shall be designed for 6, 7–8, 9–11, 12–17, 18–34. 35 teeth and over.

15 The number of teeth on which each cutter shall be based is, $M = \frac{2\,N\,n}{N + n}$, where N and n are the maximum and minimum number of teeth to be cut by any given cutter. A sprocket having N teeth will thus have a pressure angle which departs the same amount from the desired pressure angle as one having n teeth.

16 The values of M for the various cutters are respectively, 6, 7.47, 9.90, 14.07, 23.54 and 65.42 teeth.

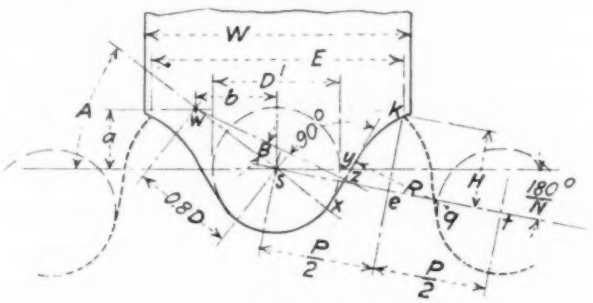


FIG. 2

17 Cutter numbers shall be respectively, 6, 7½, 10, 14, 24 and 65.

$$D' = 1.005\,D + 0.003 \text{ in.}$$

$$A = 31^\circ + \frac{90^\circ}{M}$$

$$ws = 0.8\,D$$

$$B = 13^\circ - \frac{18^\circ}{M}$$

$$xy = 1.3\,D \sin\left(25^\circ - \frac{90^\circ}{M}\right)$$

$$se = \tfrac{1}{2}\,P$$

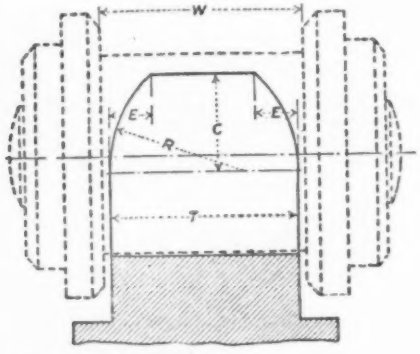
$$ek = H = 0.35\,P$$

xz is a circular arc with w as the center
zy is a straight line perpendicular to wz
yq is a line parallel to wz
q is a point so located that the arc yk, struck from q as a center, will pass through k

The radius R may be determined graphically.

18 Where the same roller diameter is used on chains of two different pitches (as $\frac{5}{8}$-in. roller, 1-in. pitch; and $\frac{5}{8}$-in. roller, $1\frac{1}{4}$-in. pitch), cutters Nos. 6 and $7\frac{1}{2}$ shall be designed for the longer pitch, and may be used for cutting sprockets of the shorter pitch.

TABLE 2 DIMENSIONS OF ROLLER TRANSMISSION CHAIN SPROCKETS



R (Min) = 0.43 P. W = Chain width.

Pitch (P)	Sprocket Width (T)	C	E	Sprocket Bottom Diam. Minus Tolerance	Nominal Clearance (W − T)
$\frac{3}{8}$	0.227 ± 0.004	0.113	$\frac{3}{64}$	0.004	0.023
$\frac{1}{2}$	0.284 ± 0.005	0.150	$\frac{1}{16}$	0.006	0.028
$\frac{5}{8}$	0.343 ± 0.006	0.1875	$\frac{5}{64}$	0.007	0.032
$\frac{3}{4}$	0.459 ± 0.007	0.225	$\frac{3}{32}$	0.009	0.041
1	0.575 ± 0.008	0.300	$\frac{1}{8}$	0.012	0.050
$1\frac{1}{2}$	0.692 ± 0.010	0.375	$\frac{5}{32}$	0.015	0.058
$1\frac{1}{2}$	0.924 ± 0.012	0.450	$\frac{1}{16}$	0.018	0.076
$1\frac{1}{4}$	1.040 ± 0.013	0.525	$\frac{7}{32}$	0.021	0.085
2	1.156 ± 0.015	0.600	$\frac{1}{4}$	0.024	0.094
$2\frac{1}{2}$	1.447 ± 0.018	0.750	$\frac{5}{16}$	0.030	0.115
3	1.738 ± 0.021	0.900	$\frac{3}{8}$	0.036	0.137
4	2.319 ± 0.027	1.200	$\frac{1}{2}$	0.048	0.181
5	2.784 ± 0.032	1.500	$\frac{5}{8}$	0.060	0.216
For-mulas	T = 0.93W − 0.006 Tolerances = ± (0.01W + 0.002	0.3P	P/8	0.012P	0.07W + 0.006

All dimensions in inches

Cutters Nos. 10, 14, 24 and 65 shall be designed for the shorter pitch and may be used to cut sprockets of the longer pitch.

19 W = width of cutter = $1.02 \times O.\,D. \times \sin \frac{180^\circ}{n}$,

to the next higher 32d inch; where $O.\,D.$ equals the outside diameter of the sprocket with the larger pitch and the lowest number of teeth to be cut with the cutter.

NOTE: The specifications to the cutter maker may be simplified if w and z are located as follows:

$$a = 0.8 \times D \times \sin\left(31^\circ + \frac{90^\circ}{M}\right)$$

$$b = 0.8 \times D \times \cos\left(31^\circ + \frac{90^\circ}{M}\right)$$

$$xz = 2.6 \times D \times \sin\left(6.5^\circ - \frac{9^\circ}{M}\right)$$

20 This proposed standard tooth form and the cutters which produce it have slight modifications from those in general use, but the principles involved have all been thoroughly tried out and the compromises that have been made to produce a standard are in no case large enough to influence the result. Sprocket teeth cut to similar forms have been in very general use for a number of years.

21 Standard chains made to fit the old form of sprocket teeth will fit the new form as well, but they will run better on the new sprockets.

STANDARD UNIFORM SYSTEM FOR NUMBERING CHAINS

22 In this system of numbering, each chain number consists of three parts, the first indicated by the first one or two digits; the second by the last digit, and the third part by the letter which follows the figures. The rule for fixing the number of any standard chain is as follows:

- *a* The left-hand figures shall denote the number of one-eighth ($\frac{1}{8}$) inches in the pitch
- *b* The final figure shall denote the series to which the chain belongs. "0" stands for the heavy series, "1" for the medium series, and "2" for the light series
- *c* The letters "*W*" and "*N*" following the figures shall denote whether the chain is wide or narrow.

Thus, if the pitch is $\frac{3}{8}$ in. and the chains is a wide one of the heavy series, its number is 30*W*. Again, if its pitch is $1\frac{3}{4}$ in. and it is a narrow chain of the medium series, the standard number is 141*N*.

23 To provide for rollerless bushing chains the committee proposes the use of the digits 5, 6 and 7 in place of 0, 1 and 2.

No. 1833

REPORT OF RESEARCH SUB-COMMITTEE ON LUBRICATION

THE Sub-Committee on Lubrication has continued its study of lubrication problems during the past year, keeping in touch also with investigations being carried on elsewhere so far as these have any direct bearing on the fundamental questions involved in explaining the action of lubricants.

2 In the previous report, which was presented at the Detroit meeting of the Society in 1919,[1] some mention was made of experimental results already secured by the members of the sub-committee, and attention was called to a number of problems which seemed of particular significance.

3 In connection with one of these problems, namely, that of determining change of viscosity of lubricants under high pressure, the following report of progress may be submitted.

4 Several oils of diverse character have been investigated up to a moderately high pressure with both high and low temperature, as shown by Table 1. It will be noticed not only that the percentage increase of viscosity over and above its initial value at the temperature in question due to an application of 6000 lb. per sq. in. pressure is quite pronounced at both temperatures, but that the relative sequence of the different oils with respect to the pressure effect is not the same at the temperature of steam (100 deg. cent.) as it was found to be at ordinary temperature (25 deg. cent.). Thus at 25 deg. cent. one of the mineral oils (Texaco) showed a very much larger pressure coefficient than lard oil, the numerical change of viscosity of the Texaco having been 290 per cent as against 75 per cent for lard oil. On the other hand, when these same two

[1] MECHANICAL ENGINEERING, vol. 41, June 1919, p. 537.

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samples of oil are subjected to the same change of pressure at 100 deg. cent. the lard oil now shows the greater effect, namely, 34 per cent as against only 31 per cent for the Texaco. The Committee considers this fact of the influence of temperature on the relative

TABLE 1 CHANGE OF VISCOSITY DUE TO A PRESSURE OF 6000 LB. PER SQ IN. (ABOUT 400 KG PER SQ. CM.)

Oil	Approx. Abs. Viscosity (C.G.S.) for Atmos. Pressure at 25 Deg. Cent.	Approx. Percentage Increase of Viscosity	
		At 25 deg. cent.	At 100 deg. cent.
Lard	0.5	75	34
Sperm	0.4	72	29
Asphalt base (Texaco)	1.5	200	31
Paraffin base (Veedol)	0.9	160	50

behavior of a series of oils to be a significant one not previously brought forward.

5 Recently the pressure range available for these experiments has been increased to 3000 kg. per sq. cm. (44,000 lb. per sq. in.) and a series of different lubricating oils are being studied

TABLE 2 APPROXIMATE RESULTS OBTAINED ON VEEDOL AND LARD OIL

Effect	Veedol	Lard Oil
Solidifying pressure at about 21 deg. cent.	1240 kg. per sq. cm. (18,200 lb. per sq. in.)	1550 kg. per sq. cm. (22,800 lb. per sq. in.)
Increase in viscosity at 100 deg. for 3000 kg. per sq. cm. (44,000 lb. per sq. in.)	1900 per cent	600 per cent
Increase in viscosity at 100 deg. for 1500 kg. per sq cm. (22,000 lb. per sq. in.)	375 per cent	240 per cent

over this range at both room temperature and 100 deg. cent. The various samples all show curves strongly concave upward when viscosity is plotted as ordinate against pressure as abscissa, with a tendency for a sharp increase which may practically be called

solidification at some fairly definite pressure, but the individual oils show considerable variation. Table 2 shows approximate results obtained on Veedol and lard oil, which will be subject to very slight numerical corrections when the apparatus is completely standardized.

6 The object of investigating the change of viscosity under pressure is primarily, as indicated in the report presented at the Detroit meeting,[1] to discover whether this phenomenon has any bearing on the fact that certain oils, notably lard oil, are more effective than other lubricants under some conditions. A secondary object is to map out the complete empirical curves characterizing the viscosity of the most important lubricating oils as functions of pressure and temperature. The viscosity-temperature curves at atmospheric pressure form a recognized factor in oil specifications, but such data have not been available at higher pressures prior to the work of this Committee.[2]

7 The investigation of viscosity under pressure is being carried out by Mr. M. D. Hersey of the Sub-Committee, at the Massachusetts Institute of Technology. The Committee wishes to acknowledge the courtesy and coöperation of the Bureau of Standards and the Jefferson Physical Laboratory of Harvard University in important features of the work. The apparatus in use is a further modification of that previously described[3] and is based on the rolling-

[1] A. S. M. E. Spring Meeting, Detroit, Mich., June, 1919.

[2] The problem of measuring viscosity of lubricants under pressure has gone forward in three steps as follows: First, this Committee discovered the existence of the effect and made quantitative determinations on one fixed oil and one mineral oil at 20 deg. cent. over a pressure range of 200 kg. per sq. cm. (3000 lb. per sq. in.), these data having been published by M. D. Hersey in 1916 (see footnote [3]). Second, the pressure-viscosity curves for four fixed and four mineral oils were investigated at the National Physical Laboratory (England) over a pressure range of about 1200 kg. per sq. cm. (18,000 lb. per sq. in.) the temperature in all cases being 40 deg. cent. (J. H. Hyde, Proc. Roy. Soc. A, vol. 97, pp. 240–259, 1920; Report of Lubricants and Lubrication Inquiry Committee, Department of Scientific & Industrial Research, London, pp. 80–90, 1920.) This important investigation corroborated and greatly extended the previous work, showing that at 40 deg. cent. the pressure effect on fixed oils taken as a class is less than on mineral oils. Third, as will be seen from the present report, the factor of temperature variation has been introduced, at the same time increasing the pressure range to 2½ times that of the British tests referred to above.

[3] *Jour. Wash. Acad. Sci.*, vol. 6, pp. 525–530, 1916; *Mechanical Engineering*, vol. 41, June, 1919, p. 537.

ball principle developed by Mr. A. E. Flowers of the Sub-Committee. It is expected that a more complete account of the work can be prepared for publication in the near future.

ALBERT KINGSBURY, *Chairman*
ALAN E. FLOWERS
MAYO D. HERSEY
Sub-Committee on Lubrication.

No. 1834

REPORT OF SUB-COMMITTEE ON BEARING METALS

RESEARCH COMMITTEE OF THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

IN the January number of MECHANICAL ENGINEERING our Committee made a brief report giving concisely the reason for abandoning our original plan of first publishing a complete compendium of the technical literature on bearing metals. At the time, we also pointed out what we considered the most promising outlook for future study and investigation.

2 It is our conviction that a large amount of valuable time is being wasted in testing bearing alloys, in determining characteristics that are of no consequence and are utterly meaningless. The only mechanical or physical test of a bearing alloy, in the aggregate, that can serve any useful purpose, is that of determining whether the given alloy has a sufficiently high factor of safety against possible distortion or rupture under specified service conditions at service temperature. All other characteristics are of a microscopic nature, depending entirely upon the properties of the individual crystals of the alloy, upon their orientation, relative hardness, fusing temperature, quantitative proportions and the like.

3 Our studies have convinced us that any work on our subject without the application of modern metallography is an absolute waste of time; we also feel that studying the hardness of the individual microscopic crystals is of the greatest importance in supplementing the application of metallography to our subject.

4 The instrument shown in our last report has been perfected to a much higher degree and three of them have been substantially completed, one for each member of our Committee. This instrument determines the characteristic of a crystal which is the combination of three of the five fundamental conceptions of hardness: [1]

[1] General Metallurgy — Hofman.

Received by the Council and ordered printed. Presented to THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS at the Spring Meeting, June 1919.

namely, the combined effect of cutting hardness, scratch hardness and penetration hardness. This is done by very slowly moving an exceedingly hard and sharp point[1] under a definite pressure, over a highly polished surface of the crystal to be tested; in fact, the point is moved so slowly that no additional penetration is effected by stopping. The point is a solid right angle or the corner of a cube, mounted in such a manner that the diagonal of the cube will be normal to the surface tested, and having an edge of the cube advance directly in the line of motion. The motion is effected by a slow micrometer feed. The width of the cut, scratch, or penetration is a function of its depth and, therefore, measuring the width of the mark gives us the means for determining this combined characteristic representing the hardness of any crystal. The instrument is properly called a *Microcharacter*.[2]

5 In order to lay the broadest foundation for our work, one that will facilitate the coöperation of other investigators and will make all results directly comparable, it will be necessary that conventional units be established; that is, that a *Scale of Microhardness* be determined upon. With the establishment of this scale, it will be possible to consider the various crystals in the many different alloys in terms of exact degrees of hardness. This is the work that occupies the attention of your Committee at the present time.

6 Experience well shows that a single homogeneous metal is not suited for bearing purposes, and that the first requirement for a bearing metal is that it be an alloy composed of at least two metals, or a metal and a metalloid, which shall have at least a limited degree of solubility while in the molten state; but that upon cooling it shall partially separate out into dissimilar crystals, and thus form the proper microstructure which is necessary in all bearing alloys. It is not only necessary that a bearing alloy shall be composed of chemically dissimilar crystals, but it is all-important that these crystals shall have a marked degree of difference in their physical hardness and wearing qualities.

7 The degree of relative solubility necessary for the constituents of bearing alloys is well illustrated by the copper-tin and the copper-zinc compounds, and this also illustrates why bronze is a better bearing alloy than brass. For example, with the addition of 11 per cent of tin to copper, a three-phase alloy is ordinarily produced — less than a two-phase alloy cannot be produced; even

[1] MECHANICAL ENGINEERING, p. 71, January, 1919.

[2] μικρός, small; χαράσσειν, to engrave, or scratch.

with the most instantaneous chilling effect, crystals of different chemical composition are produced. While on the other hand, with 11 per cent of zinc added to copper, only a single-phase alloy can be produced, however slowly it may be cooled; that is, all of the crystals in this copper-zinc alloy are of the same chemical composition and, therefore, all have the same physical properties. Quite the opposite is true in the copper-tin alloy, an extreme dissimilarity of crystals existing both as to chemical composition and physical hardness. Now, it is due entirely to this dissimilarity of crystals in the bronze that makes it a better bearing alloy than the brass.

8 In the solidifying of the molten bronze the tendency of the tin to separate from the copper is far greater than that of the zinc in the solidifying of brass, for the reason that the bronze solidifies with a distinct microscopic heterogeneity. The matter of having a proper microstructure in a bearing alloy is always of far greater importance than its exact aggregate chemical composition. The value of an exact or definite chemical composition is secondary, in that it can serve only in producing the desired microstructure in a given alloy, provided the alloy is subjected to the proper cooling conditions.

9 The essential characteristic of all bearing alloys is a structure made up of alternately hard and relatively soft microscopic particles intimately mixed. The function of the hard particles or *bearing crystals* is to support the load and resist the wear. These bearing crystals should not be hard enough to prove distinctly abrasive to the journal surface. General experience shows that an extreme hardness of the bearing crystals is characterized by an excessive wear of the journal. The function of the softer or more readily abraded crystals is that of being plastic and permitting the bearing crystals to adjust themselves to surface requirements of the journal. These softer crystals are also more readily abraded, and therefore wear slightly below the surface of the bearing crystals and thus form slight depressions upon the bearing surface which serve for the retention of the lubricant. However infinitesimal in amount this may seem, nevertheless it is this lubricant that prevents scoring or seizing when the journal is starting up from rest at a time when actual metallic contact between the bearing surfaces exists. The same is equally true under an excessive load. This function of retaining a slight quantity of the lubricant upon the bearing surfaces when metallic contact exists, characterizes a bearing alloy in its truest sense. Therefore a bearing metal may be

defined as: *An alloy that is capable of retaining a lubricant on a bearing surface.*

10 In the operation of a bearing under normal conditions, when a continuous and unbroken film of lubricant exists it matters little what metals are used while the film is sustaining the entire load. In the starting and stopping of the journal, however, or at all such other times when the film is interrupted and metallic contact exists, it then becomes very important that the properties of a bearing metal should be present.

11 A matter of importance, which seems not to have been considered heretofore, is the fusing temperature of the bearing crystals. From observations made it is evident that under severe conditions where relatively low-fusing bearing crystals exist in a high-fusing alloy these bearing crystals actually fuse on their surfaces during the process of the "running in" of the bearing. The delta tin-copper crystal ($SnCu_4$) may be cited as a particular example, and in some severe conditions corresponding to automotive worm-drive service the alpha copper-phosphorus crystal (Cu_3P) seems to function in a similar manner. It is doubtful whether these conditions ever obtain in any of the babbitts, since their bearing crystals are the highest-fusing compounds of these alloys.

12 The study of bearing metals is very incomplete unless these alloys are considered in conjunction with the other corresponding bearing member. The extreme variety of modern steels makes this necessary, since a bearing alloy suited for a soft low-carbon machine-steel journal would not be an economic selection for a high-carbon nickel-chrome heat-treated journal, and a most economic selection for the latter would prove destructive to the former.

13 It seems exceedingly improbable that laboratory accelerated service tests can ever give general satisfaction, owing to the difficulty of reproducing in a few hours' time the equivalent of many years of service conditions. It is our conviction that much more can be learned from the study of failures, and also in studying old bearings together with their journals, which have given eminently satisfactory service for an exceptionally long period.

C. H. Bierbaum, *Chairman*
J. A. Capp
H. Diederichs
Sub-Committee on Bearing Metals



LESTER GRAY FRENCH

No. 1835

NECROLOGY

LESTER GRAY FRENCH

Lester Gray French, editor and manager of MECHANICAL ENGINEERING, and for thirteen years editor of the publications of the Society and latterly assistant secretary, died on April 18 at the French Hospital, New York City, from septic poisoning following an operation.

Mr. French was born in Keene, N. H., in April 1869. He received his technical education at the Massachusetts Institute of Technology, from which he was graduated in 1891 with the degree of S.B. He served the year 1891–92 with the Cranston Printing Press Company as draftsman, and then became connected with the International Correspondence Schools, Scranton, Pa., as instructor in mechanical engineering. Leaving this institution in 1895 he went to Providence, R. I., where he entered the employment of the Builders Iron Foundry as assistant to the superintendent. In 1897 he became editor in chief of *Machinery*, continuing in this position until 1906, when he resigned to take up the publication of technical books, among them being one of the earliest American treatises on the steam turbine, of which he was the author. In 1908 he was made editor of the publications of the Society, serving faithfully and ably to the last in that capacity.

Particulars regarding his activities at Scranton and on *Machinery* are given in the tributes printed below from his associates, Professor Goodenough and Messrs. O'Neill, Rogers and Flanders.

Mr. French was elected a junior of the Society in 1899 and became a member in 1912. He was also a member of the Engineers' Club of New York City, and of the University Club of White Plains, N. Y., where he resided.

Resolutions of appreciation of the work of Mr. French were adopted by the Council of the Society and by the Boiler Code Committee at their April meetings, as follows:

Resolved: That the Council of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS hereby expresses its deep regret and sorrow at the loss by death of Mr. Lester Gray French, esteemed Member of the Society since 1899 and for the past fourteen years Editor of its Journal and other publications, acting also as Assistant Secretary.

With rare ability, good judgment and devotion, Mr. French has contributed in an important way to the growth and development of the Society. Not only our own Members, but the Engineering Profession, owe to him a debt of gratitude.

The Boiler Code Committee learns with great regret that Lester G. French, the assistant secretary and editor of the Society's publications, has passed away after protracted illness.

The publications have been brought to a high standard under his direction, enhancing the work and prestige of the Society. Too much credit cannot be given Mr. French for this major activity of the Society, notwithstanding that it is understood that all work of the Society is directed by Committees. Mr. French, by a rare combination of enterprise, vision, tact and devo-

tion has been personally responsible for much of the Society's progress. Mr. French has been a strong supporter of all standardization movements, particularly the Boiler Code movement, and his loyal coöperation in the work of the Committee has been thoroughly appreciated. Be it therefore

Resolved: That the Boiler Code Committee acknowledge and record their appreciation of the life work of Lester G. French, and be it further

Resolved: That a copy of these resolutions be sent to the family and also the Secretary of our Society.

The following letters of appreciation of the professional work of Mr. French and of his sterling personal qualities have been received from members and other friends well fitted by long acquaintance to bring them into review:

LETTERS OF APPRECIATION

When it was perceived by some of us that the Society ought to develop its Journal into something like its present state and the problem of a suitable man to conduct its editorial work arose, I happened to discuss with Mr. French some features of our problem, not thinking he would be available, for I knew that he had plans for the future that I supposed would be more attractive to him. To my surprise, he was interested, and to my gratification he declared that to give up his plans and undertake the work would be very agreeable to him.

Neither of us, as I remember, mentioned the subject of compensation — his interest in the matter was entirely from the standpoint of the service which he saw the prospect of rendering to the profession in helping to develop such a journal of engineering as we had in mind and which he agreed ought to be published and could be best published by a great engineering society.

Though the Society had not at the time achieved greatness, measured by our present standards, he saw that it could achieve it, and his greatest wish and ambition were to help in that work.

Bringing to the service the experience and knowledge based upon successful and high-class technical editorial work, he, with rare ability, tactfulness and complete absorption and devotion, has built for himself a monument that will endure so long as there are engineers having an appreciative understanding of what such work as he did really means.

By nature, retiring and unobtrusive — conservative, yet not afraid of new ideas, nor of departures from the beaten path when they seemed justified, he was peculiarly fitted for the task he undertook to perform; his one failing being that he could not be prevented from driving himself beyond his strength.

He had the faculty of attaching to himself and holding other capable and devoted workers and has left behind him an organization that can continue the work and develop it to the fullest degree; but a very large share of whatever success THE JOURNAL may hereafter attain will be due to the splendid foundation he did so much to construct.

As his personal friend, acquainted with him since the time when he was a student at "Tech," and as an officer of the Society, in close touch with his work, I find my pen inadequate to fully express my admiration and regard for him, and I feel sure that so long as the Society can attach to itself such men as Mr. French it will continue to grow in power and influence.

FRED J. MILLER

The life work of Mr. French was such that every one is anxious to give him credit for the wonderful foundation he laid for the publications of the Society, especially MECHANICAL ENGINEERING.

His special qualifications as an editor included the highest ideals, vision and executive ability, and his permanent contribution to the Society was the realization of an ambition for the Society's publications in coöperation with the Publication Committee, and the building of an organization to carry on the work.

Mr. French's inspiration thus lives on to guide the work of the Society, and all will take great satisfaction in trying to make the publications of the Society attain Mr. French's ambition.

CALVIN W. RICE

It has been my pleasure to have known Mr. Lester Gray French for many years. His clear perception of engineering problems, his art as a writer, and his ability as an editor placed him in the front rank of engineering editors. While it was the professional side that brought us together, it is the fine personal qualities that stand out conspicuously now as I think of the many pleasurable

occasions when we were brought together, either by painstaking suggestions for a revision of some of my contributions or in the general conferences connected with our society conventions. It was always a genuine pleasure and inspiration to meet him. JAMES HARTNESS

French gave what was best in him to our Society, which he served long and well. He was straightforward and upright and always ready to lend a helping hand. He will be remembered by all of his associates as a true and sincere friend. D. S. JACOBUS

The services rendered by Mr. Lester G. French to THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS have been greatly appreciated by those who have been close enough to know of his work. His first thought was always for the good of the Society. He was most diligent, intelligent, and conscientious. The results of his efforts to maintain the publications of the Society in volume and quality will be of lasting benefit. CHARLES T. MAIN

As an ideal companion, a staunch friend, a true engineer and a remarkable editor, Lester Gray French will remain long in the hearts and memories of his friends and associates in his future-building work for the Society. GEORGE A. ORROK

In the passing of Lester G. French THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS has lost one of the most faithful and efficient servants and many of us have lost a good friend and a pleasant companion. To those who have been in close contact with the work of the Society at its headquarters it is not necessary to dwell upon the importance of the work performed by Mr. French, but to those whose geographical situations make this contact difficult it may not be amiss to remind them of the extent to which a society of this kind is indebted to faithful workers, such as Mr. French, for the growth and prestige of the organization. Presidents, vice-presidents, and managers come and go, they establish policies and change methods of procedure. But no organization of this kind can grow and prosper that cannot command the services of such men as Mr. French, who by faithful and painstaking labor bring the policies of the management into full effectiveness.

It was my great pleasure during the several years I have served upon the Meetings and Program Committee to work very closely with Mr. French and I came to like and admire him greatly. His patience, enthusiasm and good judgment in the difficult task of procuring and selecting good papers for the Society did much to make service upon this Committee a real pleasure, and all who have had the experience of serving in any way in connection with the publications of the Society will join me in recording my profound regret that his usefulness and sunny influence have been brought to an untimely end. It is not too much to say that the present high standards of the publications of the Society and the success that has attended these publications financially are due in no small measure to his energy and good judgment.

The Society has lost a faithful servant, the world has lost a fine high-minded citizen, and many of us mourn the loss of a good friend, whose place will be difficult to fill.

DEXTER S. KIMBALL

During the summer of 1893 Mr. French was one of a group of four young college graduates occupying a room in the Coal Exchange Building, Scranton, Pa. These men were engaged in writing the original instruction papers for the Correspondence School of Mechanics, as it was then termed. Mr. French wrote several of the papers, his special contribution being the Applied Mechanics. Toward the close of the year he was transferred to another office and put in charge of the clerical force that examined students' work.

The papers written by Mr. French were certainly among the best that have been published by the International Correspondence Schools. They showed on the part of the author excellent grasp of the subject-matter, fine judgment in the selection and arrangement of topics, and power of clear exposition. This early experience in a rather unique field of technical writing was undoubtedly a good preparation for Mr. French's subsequent work as an editor.

The high character of Mr. French's work gained him the respect of his associates; his excellent social qualities gained their warm and lasting friendship. In his relations with his associates Mr. French was uniformly courteous and considerate; he was ever ready with suggestions or offers of assistance. Outside of the office he was the ideal companion for an excursion to the hills or a social evening.

The short summer passed too quickly. The group was broken up never to be reunited. But in the minds of each the memory of the association with Mr. French will ever remain.

G. A. GOODENOUGH

Lester G. French was editor of *Machinery* from 1897 to 1906, a nine-year period during which the machine-tool industry of this country was laying the solid foundations for the rapid development that was to follow. By education, extended study and practical shop experience, Mr. French was well equipped for the editorship of a technical journal in the metal-working field and rendered valuable service in that capacity. He had a natural talent for explaining mechanical principles and describing shop practice, especially for the practical man at work in the industries, who wants his information and data direct and hasn't the time, even if he has the taste, for fine writing. Mr. French knew his subjects and wrote with clearness and accuracy — valuable attributes in a writer whose work is essentially educational. A man of sound character, an associate it was a pleasure to work with.

M. J. O'NEILL

My acquaintance and association with Lester Gray French began in 1899 on joining the editorial staff of *Machinery*. At that time the trade and technical journals presented a variety of industrial and engineering topics before their readers; hardly one specialized as do so many now. We were then passing through an era of change in the industrial world when the importance of the machine shop was just beginning to be recognized and the upbuilding of our present great machine-tool industry was in the early stages. Although personally more interested in power engineering, Mr. French recognized the desirability of specializing on machine-shop practice, machine-tool design and development, and initiated a policy that has been successfully followed by those who came after him in the management of *Machinery*.

Mr. French was a man who endeared himself to his associates. Modest, unassuming and possessed of unusual ability, he rarely asserted his opinions in contradiction of others, but by the force of example and ability won the respect of those with whom he worked. It was with sincere regret that our business association was dissolved when I took on the burden of editorial management of *Machinery* in 1906.

The development of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS JOURNAL under his expert management was in large measure due to editorial foresight and wisdom. The JOURNAL, in my opinion, stands preëminent among engineering publications. Variety of engineering interests today makes it difficult to edit a publication that will carry universal appeal to engineers of so many affiliations as represented in THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS membership, but Lester Gray French succeeded to a most notable degree. It is with profound sorrow that I write this brief tribute to his personal qualities and editorial ability.

FRED E. ROGERS

Lester French was a friend to me for seventeen years. First as my chief on the editorial staff of *Machinery*, later as a roommate in a house in the west 80's, and finally in the close coöperation of the work of the Society our lines have been in contact throughout that period.

His was a big heart and a high courage in a small body. Untiring energy and absolute honesty of workmanship were his also. He lived and spent himself for the Society. Its position in the professional world today is due in no small measure to his faithful and intelligent administration of the charge laid upon him.

It would be impossible to express in words what was the real beauty of his character; but those who knew him intimately will love best to remember the health and purity of his sentiments. It is not that he was sentimental; but for children, for home, for family, for friends, for country scenes, for all relations and situations that reach to the roots of our common human nature, he kept an open channel to his heart. To have really known him is to have enriched one's life to its end.

RALPH E. FLANDERS

Lester French and I were classmates in college, taking the same course and becoming members of the same college fraternity. Later we were near neighbors and for some time lived together. This close personal relationship gave an opportunity for knowing the true man hidden away under his New England reserve and extreme modesty.

Capable and efficient, a quick and a hard worker in spite of ill health for years, he accomplished much in his profession. At the same time he was always interested in those things that make for the general good of the community. A man of high ideals, fond of music, art and literature, gifted with a keen sense of humor, always optimistic, he was the most interesting of companions and the truest and most loyal of friends. A more thoughtful and unselfish man in his home relations could not be found. It was his delight to anticipate every desire of those dear to him and to make the home circle one of constant happiness.

Proud we are of his life and his courageous end; happy, too, in the memory of all the joy he has brought into our lives and of the example he has given us of a life well lived.

CHARLES W. AIKEN

No activity of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS so directly reaches the great body of its membership as does its publishing business.

Few of the Society's membership know to what a large extent the efficient conduct of its publications has been due to the faithful and efficient service which Mr. French, as the Society's editor, has so long rendered. Under his leadership the Society's monthly journal has been transformed from a barren reprint of papers read at the meetings to a great, influential technical journal.

Mr. French's task has been no easy one. The editor of a publication carried on by a society has to serve many masters, and the masters change year by year. Under these circumstances, to maintain a continuity of policy and of development — and these are necessary to success — calls for rare patience, tact and wisdom.

We all owe a debt to this quiet, modest man whose use of his talents in the service of his profession has been a large factor in its upbuilding, and whose fine personality won for him a wide circle of friends.

CHARLES WHITING BAKER

I knew Mr. French from the time he first entered the editorial service of *Machinery*. It was my privilege to have been in close contact with him in editorial work upon which I was engaged. The most impressive fact about Mr. French to me was his invariable honesty — honesty to himself, to his ideals, and honesty in his work. This loss is a personal one to me which I feel keenly. The loss to the mechanical engineering profession is greater than anyone knows, even those who were closest to him.

The publication MECHANICAL ENGINEERING is L. G. French's monument, a living monument, and a wonderful one. I regard the success of this publication as being due to him more than to any other individual.

I hope the members of the Society will think of him with pride whenever they see this journal.

G. M. BASFORD

The life of Lester G. French was a refutation of the theory that an engineer does not concern himself with the human element. Accessible, cordial, never too busy to give his attention to any demand for coöperation or help, his personality was an important factor in the spirit of hospitality of the Engineers' Building. Forgetting himself, he went through the many months of ill health with the spirit of Stevenson. His contributions to engineering through his many years of devotion to the JOURNAL OF THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS, where the best of engineering theory and practice is recorded, insure a high place for him in the history of engineering achievement, but his warm-hearted interest in humanity and its advancement mean more than this and mark his achievement as a giver of happiness minutes to the world.

FRANK B. GILBRETH

My own experience with Mr. French leaves a very happy impression in my mind. He always seemed to have the interest of the Society at heart and to be high-minded about it. I went into his office many times and always came away with a feeling of having been helped. It will be very difficult to replace him.

IRA N. HOLLIS.

The Society has lost a splendid servant, the engineering profession one of its outstanding spokesmen and all of us a loyal friend. We have experienced a very real loss.

MORRIS L. COOKE

The Society has lost a very useful and efficient member who had its interests at heart at all times and with whom it was a great pleasure for his associates to work.

FREDERICK A. SCHEFFLER

All this he was — and more — to those whose privilege it was to be in daily association with him in his work, and who, having shared in his clean enthusiasms and experienced the unfailing kindliness and consideration it was ever his nature to bestow, will hold him in affectionate and enduring memory.

L. E. MAURICE ADAM

L. E. Maurice Adam, administrator of Ingenio Valdez, Ecuadora, the largest sugar factory in Ecuadora, died on October 24, 1921. Mr. Adam was born

on October 23, 1892, in Curepipe, Isle of Mauritius. He was graduated from the Royal College of Mauritius and from Louisiana State University with the degree of B.S.

He served at one time as superintendent and chief chemist of the Catherine Plantation, Louisiana, becoming, in 1916, chief chemist with the Cuba Cane Sugar Corporation, Camaguey, Cuba. He resigned from the position of assistant manager with this firm to become administrator in the largest sugar factory in Ecuadora where he was engaged in the installation of modern methods and machinery at the time of his death.

Mr. Adam became a junior member of the Society in 1917. He received in 1916 THE AMERICAN SOCIETY MECHANICAL ENGINEERS Student Prize for his paper on The Adaptability of the Internal-Combustion Engine to Sugar Factories and Estates. He was also a member of the American Chemical Society and of several clubs and fraternal organizations.

GUSTAF AKERLAND

Gustaf Akerlund, senior member of the firm of Akerlund and Semmes Inc., 17 Battery Place, New York City, and a member of the Society since 1913, died on December 10, 1921. Born in Sweden in November 1879, Mr. Akerlund received his education there, graduating from the Boras School of Technology in 1901 with the degree of M.E. He served his apprenticeship during summer vacations with the Boras Machine Works, and for a year after his graduation was employed as a designer by the Moritz-Jhar-Machinen Fabrik Gera Reusse in Germany.

In 1902 he came to the United States. His first position here was in drafting and designing in the boiler department of Babcock and Wilcox, Bayonne, N. J. From there he went to Philadelphia to the gas-producer department of R. D. Wood and Co. Later he did special work in gas-producer testing at the U. S. Coal Testing Station in St. Louis. For three years Mr. Akerlund was in charge of the experimental department of testing gas-power plants for the Westinghouse Machine Co. of East Pittsburgh, after which he built and tested a gas producer of his own design. Previous to the formation of Akerlund and Semmes, Inc., he was chief engineer for the Standard Gas Power Co. of Atlanta and the New England Gas Producer Co. of Boston.

FRANCIS B. ALLEN

Francis B. Allen, vice-president of the Hartford Steam Boiler Inspection and Insurance Co., Hartford, Conn., died on July 27, 1921. Mr. Allen was born on June 1, 1841, in Baltimore, Md. He received a common school education and in 1862 was appointed to the Engineer Corps of the U. S. Navy, where he was assigned to the U.S.S. *Port Royal* as third assistant engineer.

In October of 1863 he was advanced to the position of second assistant engineer on the *Port Royal* which joined the East and West Gulf Squadrons. In 1865 Mr. Allen was promoted to the rank of Master, now junior lieutenant and assigned to the *Dictator* of the Atlantic Squadron. A year later he was ordered to New York for experimental work on the expansion of steam. These experiments were completed in 1867 when he was assigned to the U.S. Flagship

De Soto of the West Indian Squadron. During the following winter he resigned from the Navy and entered civil life.

Mr. Allen was vice-president of the Hartford Steam Boiler Inspection and Insurance Co. for a number of years. He became a member of the Society in 1880. He was a member of a number of Naval associations and vice-president of the Army and Navy Club of Connecticut. He was well known in insurance, engineering and veteran circles.

DAVID HERBERT ANDREWS

David Herbert Andrews, president of the Boston Bridge Works, Inc., East Cambridge, Mass., died on February 24, 1921. Mr. Andrews was born on September 17, 1844, in Pepperell, Mass. He was graduated in 1869 with the degree of B.S. from the Chandler School of Science and Arts, connected with Dartmouth College. In 1908 the honorary degree of M.A. was conferred upon him by that institution.

For two years he taught mechanical drawing in a night school at Fall River, Mass., then becoming connected with the National Bridge and Iron Works at East Boston, Mass., as engineer. He was the designer for that concern of the trainshed roof at the Boston Terminal for the Boston and Lowell Railroad, the building of which caused considerable interest in engineering circles at the time. The company met with financial difficulties in 1876 and its affairs were put into the hands of receivers. Mr. Andrews bought such of the tools and machinery as he needed and established his own business in East Cambridge under the name of the Boston Bridge Works. A list of the work done during the first twenty years of the company's existence would comprise most of the railroad and highway bridges of New England and a considerable number elsewhere. In 1901 the company was incorporated with Mr. Andrews as president.

Mr. Andrews was a member of the American Society of Civil Engineers, the Boston Society of Civil Engineers, the Engineers' Club of Boston and the Boston City Club. He became a member of our Society in 1913.

FRANK D. BAKER

Frank D. Baker, for twenty years chief engineer of the Colorado department of the American Smelting & Refining Co., died in Denver on April 29, 1921. Mr. Baker was born in March 1858, in Wilmington, Ill. He was graduated in 1888 from the mechanical engineering course in the University of Illinois. During his long service with the Smelter Co., Mr. Baker was in charge of, or intimately connected with, much of the metallurgical plant construction in the state and was widely known and admired by western metallurgical and mechanical engineers. He was the inventor of numerous metallurgical devices of which the best known is the Baker Cooler for cooling calcined materials on a large scale.

Mr. Baker became a member of the Society in 1893.

WILLIAM EDGAR BAKER

William E. Baker, senior member of the firm of W. E. Baker & Co., consulting engineers, New York City, died on November 7, 1921. Mr. Baker was born in Springfield, Mass., on October 18, 1856. He was graduated from La-

fayette College in 1877 with the degree of C.E. and then entered the service of the St. Paul & Pacific Railway, where he remained until 1881 when he engaged in reconnaissance work for the Canadian Pacific Railway.

In 1884 he became chief engineer of the International and G. N. Railway. Five years later Mr. Baker entered the works of the General Electric Co. at Lynn, Mass., there to acquire in a practical way the knowledge of electricity and in particular its application to the traction problem. In a few months he was made division superintendent and later he became general superintendent of the electric street railways of Boston, known as "The West End."

In 1893 he resigned from this position to represent the General Electric Co. in Chicago at the World's Fair as chief engineer and general manager of the Columbia Intermural Railway. In 1894 he became general manager and chief engineer of the Metropolitan West Side Elevated Railway, Chicago, and from 1899 to 1902 he was general superintendent of the Manhattan Railway, New York. Since 1902 Mr. Baker had been engaged in consulting work, with offices in New York City, being specially interested in the London Underground and the electrical equipment of the Calumet and Hecla copper mines.

Mr. Baker was a member of the American Society of Civil Engineers, the American Institute of Electrical Engineers and the American Museum of Natural History. He belonged to the Engineers' Club of New York and was a trustee of Lafayette College. He became a member of our Society in 1902.

RAYMOND S. BALDWIN

Raymond S. Baldwin, director of the Carbon Steel Co., Pittsburgh, Pa., died on September 3, 1921. Mr. Baldwin was born on June 22, 1881, in Newark, N. J. He was educated in Newark Academy, St. Paul's School, Concord, N. H., and Stevens Institute of Technology from which he was graduated in 1903. For one year he was in the employ of the New York Central Railroad in the mechanical engineering department and for two years was with the Consolidated Gas Co. of New York as assistant engineer in charge of the building, construction and installation of machinery.

In 1906 he became associated with the Carbon Steel Co. where he was located for five years, the last four holding the position of secretary. In 1911 he took charge of the design and construction department of J. H. Williams Co. (drop forgings), Buffalo, N. Y., later becoming manager and then vice-president and comptroller. Five years later he returned to the Carbon Steel Co. as director, resigning from the Williams Co. shortly before his death.

Mr. Baldwin became a member of the Society in 1918.

GEORGE HURLBUT BARBOUR

George H. Barbour, mechanical and structural engineer of Pittsburgh, Pa. and a member of the Society since 1912, died on April 19, 1921. Mr. Barbour was born on March 3, 1867, in Allegheny, Pa. He was educated in the Allegheny schools and the Western University of Pennsylvania, now the University of Pittsburgh, from which he was graduated as a civil engineer in 1887, one of a class of six.

Mr. Barbour's first employment was with the Schifler Bridge Co. He was

next with the Pittsburgh Testing Laboratory, serving for three years as resident engineer for that concern in Chicago, Ill., Atlanta, Ga., Philadelphia, Pa., and Buena Vista, Va. Desiring a more practical knowledge of steel manufacture Mr. Barbour entered the Lewistown Steel Works, Lewistown, Pa., where, by special arrangement, he spent one year working through every department of the mills. For the next eight or nine years he was associated with many of the best known steel and bridge manufacturing firms of the Pittsburgh district—the Carnegie Steel Co., the Keystone Bridge Co., the Fort Pitt Bridge Co., Jones & Laughlin and the Marshall Foundry & Construction Co.

From 1901 to 1917 Mr. Barbour was employed in the offices of the U. S. Steel Corporation holding successively the positions of assistant chief civil engineer, assistant to the president and assistant chief mechanical engineer. During this period he devoted his spare time to the invention, perfection and patenting of between twenty and thirty devices for rail and steel manufacture and concrete structural details.

In 1917 Mr. Barbour severed his connection with the U.S. Steel to take up special work for the Dravo Construction Co., then planning a series of enormous coal pits on Brunot's Island. That work completed, he was employed by the H. Koppers Co. during the remainder of the war period and then entered the employ of the Schaeffer Engineering and Equipment Co. which was later absorbed by the Fawcus Machine Co. with which concern he was associated at the time of his death.

ELMER A. BEAMAN

Elmer A. Beaman, until recently treasurer and general manager of the Beaman & Smith Co., machinists, Providence, R. I., died on January 4, 1921. Mr. Beaman was born in Worcester, Mass., on September 2, 1846. Much of his boyhood life was spent in the West where his family moved when he was about seven years old. In 1862 he came East and enlisted in the 23d Regiment, N. Y. State National Guard. This regiment entered the government service and Mr. Beaman fought in the Battle of Gettysburg. He was discharged from the Service in 1863 when he went to Worcester to learn the machinist's trade. In 1864 he again enlisted, this time serving in the 42d Regiment, Massachusetts Volunteer Militia, and remained in the Service until the end of the War when he returned to Worcester and finished his trade, studying evenings.

In 1866 he went to Providence, R. I., where he was employed by the Star Tool Co. of which he afterwards became treasurer. Later he founded the Beaman & Smith Co., acting as treasurer and general manager until obliged to resign on account of ill health in October of 1920.

Mr. Beaman was a 32d Mason and a Shriner. He was a member of the Providence Chamber of Commerce, past-president of the National Metal Trades Association and a member of Prescott Post No. 1, G.A.R. He became a member of our Society in 1891.

DR. GEORGE F. BLESSING

George Frederick Blessing, professor of engineering at Swarthmore College, passed away at his summer home at Bay View, Mich., June 24, after a brief illness. He was born in Carrollton, Ky., in 1875, and was a graduate of Ken-

tucky University and also of Hanover College, Ind., from which he received the degree of Ph.D. in 1908. After graduation, Professor Blessing spent several years in practical work, becoming professor of mechanical engineering at the University of Nevada in 1899, where he stayed till 1905, spending his summers in practical work with various concerns on the Pacific Coast. In 1905 he became connected with the General Electric Company at their Lynn works as a designer in the steam-turbine department, and in 1906 he became assistant professor of machine design at Cornell University. In 1908 he accepted the position at Swarthmore which he held at his death.

Professor Blessing was essentially a great teacher. Gifted with a brilliant mind, he also possessed the interest in his fellow-beings so essential to the successful teacher. At the time of his death he was employing a leave of absence from university work in making an investigation of industrial conditions for Mr. John D. Rockefeller, Jr., in the mines and mills of the Colorado Fuel and Iron Company in Colorado, and in endeavoring to develop new ideas in this work. The result of this work and similar previous experiences was to have been published in book form and it is hoped that the large amount of material gathered can be edited and published in the near future, as such a volume would undoubtedly prove of great value to students of industrial conditions. Overexertion incident to this work brought on a physical breakdown from which he failed to rally, and his illness was of short duration.

Professor Blessing was the author of books on machine drawing and descriptive geometry and of various contributions to the technical press. He was a member of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS, the Society for the Promotion of Engineering Education, the Efficiency Society, the Engineers' Club of Philadelphia, and the honorary scientific societies of Sigma Xi and Tau Beta Pi.

In his death the Society and the teaching profession lose an able engineer, a fine teacher, and a man of sterling qualities and of lovable character. His untimely end will be deeply regretted by a host of friends and former students throughout the entire country. Few men possess the attractive personal qualities that endeared him to a very large circle of acquaintances.

WILLIAM J. BRADLEY

William J. Bradley, formerly superintendent of the Rail Joint Co., Troy, N.Y., died on July 25, 1921. Mr. Bradley was born in 1860 in New York City and was educated in the city schools, afterwards taking a special course in Columbia University. He served his apprenticeship with the Quintard Iron Works. He was at one time foreman of the drafting rooms of the Standard Oil Co., then superintendent of the Troy Steel Co., the Albany Iron and Steel Works and the Continuous Rail Joint Co., later becoming associated with the Rail Joint Co. Mr. Bradley became a member of the Society in 1905.

CHARLES WALKER CADE

Charles W. Cade, factory manager of the McKinney Manufacturing Co., Pittsburgh, Pa., died on January 22, 1921, at Bellevue, Pa. Mr. Cade was born in Cambridge, Mass., on August 1, 1879. He received his early education in

the public schools of Cambridge and later was graduated from the Cambridge Latin School and the Rindge Manual Training School. He then entered the Massachusetts Institute of Technology where he received his B.S. degree in chemical engineering in 1901.

The following year he started work with the George F. Blake Manufacturing Co., East Cambridge, Mass. In 1902 he became foreman and metallurgical chemist of the Blake-Knowles Steam Pump Works and from 1904 to 1910 was superintendent of the same company. In 1911 Mr. Cade became associated with the Crown, Cork & Seal Co., Baltimore, Md., as general superintendent, which position he held until 1915 when he resigned to become factory manager of the McKinney Manufacturing Co.

Mr. Cade became a member of the Society in 1914. He was also a member of the Western Society of Engineers and belonged to the Lambskin Club and the Masonic fraternity.

MAURICE J. CANTER

Maurice J. Canter, production engineer, Midvale Steel Co., Nicetown, Pa., died in March 1921. Mr. Canter was born in Minsh, Russia, in April 1892. He was graduated from the University of Pennsylvania in 1915, receiving the degree of B.S. His first employment was with the Remington Arms Co., Ilion, N.Y., where he served his shop apprenticeship. In 1916 he was transferred to the Eddystone, Pa., plant of the company as inspector in charge of a rifle department. Later in the same year he became associated with the Midvale Steel Co. as production engineer. Mr. Canter became a junior member of the Society in 1918.

WILLIAM B. COGSWELL

William B. Cogswell, founder of the Solvay process and manager and vice-president of the Solvay Process Co., Syracuse, N. Y., died at his home in New York City on June 7, 1921. Mr. Cogswell was born in Oswego, N. Y., on September 22, 1834. He received his early education at Hamilton Academy and at a private school conducted by Professor Root of Syracuse. He completed his education at Rensselaer Polytechnic Institute, from which he was graduated in 1851.

For three years after leaving school Mr. Cogswell served as an apprentice in a machine shop in Lawrence, Mass., and was then for a few years connected with a railroad as superintendent of machinery. In 1859 he became superintendent of the Broadway Foundry, of St. Louis, and in the following year organized the firm of Sweet Brothers & Co., Syracuse, which later became the Whitman & Barnes Manufacturing Co.

During the Civil War he was rated as a mechanical engineer in the Army and later was retained by the Franklin Iron Works to superintend the construction and operation of blast furnaces in Oneida County, N. Y. From 1874 to 1879 he was in the mining business. Three years later he became interested in the manufacture of ammonia soda. He went to Europe to make a study of the soda industry and was commissioned by Solvay & Co., Belgium, to locate a plant in the United States in 1891. Through his efforts the Solvay Process Co. became famous.

Mr. Cogswell was a member of over a hundred scientific societies, social

organizations and clubs, among which may be mentioned the American Academy of Arts and Sciences, the American Institute of Mining Engineers, the American Society of Civil Engineers, the American Chemical Society, N. Y. State Chamber of Commerce, Sons of the Revolution, and the Colonial Society of America. He belonged to many social clubs in New York City and in Syracuse.

He became a member of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS in 1880 at the time of its organization and served as one of the managers until 1882.

J. WENDELL COLE

J. Wendell Cole, district manager of William Sellers & Co.'s tool and drill-grinder department and the Detroit Emery Wheel Co., died on January 18, 1921, of angina pectoris. Mr. Cole was born in Trenton Falls, N. Y., on June 6, 1842, and received his education in the schools of New York City. He served his apprenticeship in the machine trade with the old Novelty Iron Works, which was the training school of so many of the early members of the Society. His drafting experience was obtained in designing steamship machinery while in charge of construction in that office. He was also chief draftsman in the office of the chief engineer of the Hoosac Tunnel. For two years Mr. Cole was connected with the Fuel Saving Furnace Co., specializing in steam pumps and boiler feeders. It was in 1885 that Mr. Cole became connected with the William Sellers & Co., and in 1876 with the Detroit Emery Wheel Co. in the capacity of district manager.

When as a young man, Mr. Cole was agent for a pump company, he installed the first force pump in a fireboat in New York Harbor, the *John Fuller*. He was a life member of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS, joining the organization three years after its founding in 1880. He was the honorary president of the Ohio State University Student Branch Section at the time of his death.

CHARLES J. CROSBY

Charles J. Crosby, president of the Crosby Machine Works, Jonesboro, Ark., died in Springfield, Mass., on July 31, 1921. Mr. Crosby was born in Middletown, N. Y., on April 29, 1874. He was educated in the Middletown schools and later took a course in the International Correspondence School.

He worked as a machinist until 1907 when he became draftsman and designer for A. J. Matten, Denver, Col. For two years he served as foreman of the Valley Iron Works, Williamsport, Pa., and later was with the Van Norman Tool Co., Springfield, Mass., in the same capacity. For over a year Mr. Crosby was connected with the Jonesboro Machine Co. at Jonesboro, Ark., resigning in 1917 to establish his own business of the Crosby Machine Works.

Mr. Crosby became a junior member of the Society in 1919. He belonged to the Masonic fraternity.

GEORGE MASLIN DAVIS

George Maslin Davis, locomotive designer, Norfolk & Western Railway Co., Roanoke, Va., died on May 17, 1921. Mr. Davis was born on June 11, 1880, in Moorefield, W. Va. He was educated in the schools of Winston-Salem, N. C., and attended the N. C. College of Agriculture and Engineering from which he was graduated in 1901 with the degree of M.E.

Upon graduation he was employed by W. E. Martin of Winston-Salem where he served his apprenticeship in designing and drafting, later becoming supervisor. He was next connected with the Salem Iron Works, building saw mills, cotton mills, dye and sugar machinery, as designer and supervisor of the drafting department, also as estimator of contractors. From 1906 to 1911 he was with the Winston-Salem Southbound Railway Co. as draftsman and assistant engineer, making the detailed estimate of cost for the entire road. At the close of that period he became associated with the Norfolk & Western Railway Co. in the mechanical engineer's office as draftsman, locomotive and car designer. He held this position at the time of his death. Among his many successful undertakings was making the designs of a large mallet locomotive for road service and a large mountain type passenger locomotive both of which were built in the company's shops at Roanoke.

Mr. Davis was a member of the University Club of Roanoke. He became a member of our Society in 1918.

J. J. DE KINDER

J. J. deKinder, mechanical engineer and attorney-at-law, Philadelphia, Pa., died on September 11, 1919. Mr. deKinder served in the Royal Navy of Holland from 1862 to 1871, then becoming chief engineer in the merchant service of that country. He was admitted to the bar in Philadelphia in the early fifties and was engaged in legal work from that time on. He was for a time general superintendent of the Philadelphia Water Works.

Mr. deKinder was a life member of the Society, joining the organization in 1885. He was a member of the Engineers' Club of Philadelphia, serving for a year as its president. He also belonged to a number of other clubs and associations and was well known in the engineering and legal professions.

RUSSELL W. EATON

Russell W. Eaton, for thirty years agent of the Cabot Manufacturing Co., Brunswick, Me., died on March 5, 1921. Mr. Eaton was born in Readfield, Me., on November 24, 1855. He received his early education in private schools, later entering Maine State College (now the University of Maine) from which he was graduated in 1873 with the degree of C.E.

His mill-engineering experience was gained with the firms of B. B. & R. Knight, Hyde Park, Mass., and D. M. Thompson & Co., Providence, R. I. He then started his manufacturing work as assistant agent at the Ponema Mills, Taftville, Conn. In 1885 he was engaged by the Merchants' Manufacturing Co., Montreal, Canada, as manager and he was with this firm for five years when he resigned to accept the position of agent with the Cabot Manufacturing Co. in Brunswick, Me. He was in charge of the construction of the new mill for the company, remaining its agent for thirty years, also holding for many years a directorship. He retired from active service in 1920.

Mr. Eaton was a member and past-president of the New England Cotton Manufacturers' Association, contributing many papers to that organization on factory problems. He became a member of our Society in 1894.

BENJAMIN S. EDDY

Benjamin S. Eddy was born in Indianapolis, Ind., on March 10, 1892. He received his early education in the public schools of that city and later attended Purdue University where he received his B.S. in mechanical engineering in 1914. Shortly after being graduated, he established himself in business in Indianapolis under the firm name of Benjamin S. Eddy & Co., heating and ventilating contractors and engineers.

He was successfully engaged in this business when he received his commission as first lieutenant in the Engineer Officers' Reserve Corps. In September 1917, he entered the Second Officers' Training Camp at Fort Leavenworth and in December sailed for France, where he served on engineer depot duty at St. Nazaire and Marseilles. In July 1918, he attended the Engineers' School at Langres and was afterwards assigned to the Technical Board which managed the electro-mechanical matters in the A.E.F. After the signing of the Armistice he worked in the appraisal department of the Peace Commission. At the conclusion of that work he was returned to St. Nazaire and put in charge of a dredging project at Donges.

In July 1919, he sailed for home where he was honorably discharged the following month. It was at Donges that he was gassed, seemingly so slightly that he felt no effects from it until very shortly before his death, February 19, 1921. From October 1919, until his last illness he was engaged in truck farming five miles out of Sanford, Fla.

Mr. Eddy became a junior member of the Society in 1917.

WILLIAM B. EDWARDS

William B. Edwards, mechanical engineer with the S.O. & C. Co., Ansonia, Conn., died on October 11, 1921. Mr. Edwards was born in London, England, on June 19, 1851. He was educated in the schools of New Haven, Conn., later taking a special course in mechanical drawing at Yale University.

He served a four years' apprenticeship with L. E. Osborn & Co., New Haven, where he gained his general machinery experience. He then became connected with the Upson Nut Co., Unionville, Conn., where he remained for twenty years as head of the drafting department, resigning to become associated with the Brady Manufacturing Co., Brooklyn, N. Y. He remained with this company during its consolidation with the General Ordnance Co. A short time afterwards he became mechanical engineer for the S. O. & C. Co., a branch of the United Shoe Machinery Co. of Boston, Mass., which position he held at the time of his death, a period of about twenty years.

Mr. Edwards became a member of the Society in 1911. He was a member of the Board of Trade in Derby, Conn., and belonged to the Masonic fraternity.

L. A. FISCHER

L. A. Fischer, chief of the Division of Weights and Measures of the U. S. Bureau of Standards, one of the most prominent scientists of Washington and the foremost leader of the movement to promote a uniform system of weights and measures, died on July 25 at his home in Washington, D. C.

Major Fischer was born in Washington, D. C., on January 4, 1864. His activities in the field of weights and measures began early in life when he joined the staff of the old weights and measures office of the Coast and Geodetic Survey. After many years of active service in this office he took a prominent part in the establishment of its successor, the United States Bureau of Standards, in 1901, in which institution he was made chief of the Division of Weights and Measures. In addition his scientific attainments won him a world-wide reputation as one of the leading American metrologists.

For a number of years Major Fischer was regularly appointed by the President to serve on the Assay Commission. In the course of this work he personally tested the standards employed at the mint against carefully calibrated weights which were especially tested in his own laboratories at the Bureau. In 1915 he served as a member of the jury of awards at the Panama-Pacific Exposition. During the war he was commissioned major in the Ordnance Department of the United States Army and placed in immediate charge of the important section of gage design.

Major Fischer was a graduate of George Washington University. He was a member of the Washington Academy of Sciences, the American Physical Society, the Physical Society of France, a fellow of the American Association for the Advancement of Science, past-president of the Philosophical Society of Washington, secretary, since its organization, of the National Conference on Weights and Measures. He was also a member of the Cosmos Club of Washington.

Since his early youth he had been prominently identified with athletics in the District of Columbia. He was a leading oarsman in the Potomac and Analostan Boat Clubs and later became a member of the tennis teams of the Old Bachelor Tennis Club, the Dumbarton Club and the Columbia Country Club.

Major Fischer became a member of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS in 1918 and took most active interest in its work, particularly in the field of screw threads and limit gages. He was but recently elected chairmain of the Washington Section of the Society.

GEORGE J. FORAN

George Jesse Foran, manager of the condenser department of the Worthington Pump and Machinery Corporation and the associated companies of the International Steam Pump Company, died Thursday afternoon, May 12, at his home in New York City, after an illness of some weeks.

Mr. Foran was born in Boston, Mass., Jan. 22, 1862. He was graduated from the Massachusetts Institute of Technology in 1883 and in the fall of that year entered the works of The Deane Steam Pump Company, with which firm he remained until 1886. He then became associated with the George F. Blake Manufacturing Company, where he served as salesman in the engineering field for several years, later acting also as consulting engineer, with special reference to engineering design and construction of water works, condensing and air-compressor installations, and tests and investigations in New England. Upon the completion of the new Blake Works in East Cambridge he became office manager and head of the estimating and cost department, originating the cost system then installed. Later he returned to the engineering sales department.

In 1901, after the International Steam Pump Company was formed,

Mr. Foran went to New York to accept the position which he filled up to the time of his death. During the war he served upon the committees on condensing apparatus of the U. S. Shipping Board and War Industries Board and as chairman of the American Engineering Service of the Engineering Council, which handled all questions of personnel between the various departments of the U. S. Government and the four national engineering societies.

Mr. Foran was a leader in the development of high-vacuum apparatus from the very beginning, and was responsible for the design of a large number of important installations of this type in the United States. He also accomplished a large amount of original investigation work in the several fields of mechanical engineering.

Mr. Foran was not only a collector of works of art and a connoisseur of considerable attainment, but was also especially well posted on tennis and a valued contributor to various publications, all of which is illustrative of the versatility of the man.

Mr. Foran became a member of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS in 1887 and was active in its committee work, especially as chairman of the Condenser and Heater Sections of the Power Test Code Committee and as a member of the Publication Committee. By virtue of his special knowledge of numismatics he was appointed on the new committee of the Council on Design for THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS Medal. He was also a member of the Verein Deutscher Ingenieure, the American Association for the Advancement of Science, the Engineers' Clubs of New York and Boston, and the Mohawk Golf Club, Schenectady; and an associate member of the American Society of Naval Engineers.

Mr. Foran's wise counsel will be missed at the Society's headquarters, where he was a frequent visitor. His interest in the work of the Society was uniformly great, and he was always ready and willing to render assistance to the Secretary's office upon any matter of inquiry. He was particularly close to the committee activities and to the work of the New York Section, and these especially will feel the loss of his friendly interest.

WILLIAM DUNDERDALE FORBES

William Dunderdale Forbes, formerly president of the W. D. Forbes Co., New London, Conn., died on February 17, 1921. Mr. Forbes was born in Perth Amboy, N. J., on July 10, 1852. He received his early education in the public schools of Buffalo and later received his M.E. degree from the Polytechnic Institute of Zurich, Switzerland. He also took post-graduate work at the Polytechnic in Dresden, Germany.

Mr. Forbes served his apprenticeship with Weiss & Co., Zurich. He then returned to this country and became connected with the DeLamater Iron Works, New York. Later he became engineer of the H. A. Gouge Ventilating Co., New York. He resigned from this position to become foreman of the machine shop of John T. Noye & Sons Co., Buffalo, N. Y., where he was located until he became associated with the Eaton, Cole & Burnham Co., Bridgeport, Conn., as superintendent. In 1882 Mr. Forbes established the W. D. Forbes Co. in Hoboken, N. J., and thirty-three years later moved it to New London, Conn.

Mr. Forbes was a member of the council of the Society of Naval Archi-

tects and Marine Engineers, an associate member of the American Society of Naval Engineers and a life member of our Society since 1886.

ARTHUR N. FOWLER

Arthur N. Fowler, supervisor of tool and machine works, Stanley Rule & Level Co., New Britain, Conn., died on August 24, 1921. Mr. Fowler was born in Milford, Conn., on June 27, 1867. He was educated in the Milford grade and high schools, and having served his apprenticeship, became employed by the Hartford Machine Screw Co. From 1907 to 1914 he was with the Underwood Typewriter Co., Hartford, as foreman of their tool department. At the close of that period he accepted the position of supervisor of tool and machine works of the Stanley Rule & Level Co. also having charge of the drafting room. Mr. Fowler became an associate member of the Society in 1920.

SIR CHARLES DOUGLAS FOX

Sir Charles Douglas Fox, upon whom in 1900 was conferred honorary membership in THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS, died on November 13, 1921, at the age of eighty-one. Sir Douglas received his early education at the Cholmondeley School at Highgate, studied at King's College, London, and at the age of seventeen was apprenticed to his father, Sir Charles Fox. For two of these apprenticed years he acted as resident engineer in charge of work upon Witney Railway, and subsequently was with the Ramsey Railway in the same capacity. In 1863 he joined his father's firm as a partner; later his brother, Sir Francis, became a partner, and ultimately the name of the firm was changed to Sir Douglas Fox and Partners, of which firm he was the head at the time of his death.

Sir Douglas Fox's work covered a wide field, for his name was well known in connection with the construction of railways in South Africa and Australia as it was in England. Among his earlier work was the construction of the London, Brighton and South Coast Railway, the Cape Town and Wellington Railway, and the Cape Town-Wynberg Railway. With Sir James Brunlees, he acted as consulting engineer for the Mersey Tunnel. For his work in this connection Her Majesty Queen Victoria conferred the honor of knighthood upon him. He was also connected as joint consulting engineer with the construction of the Liverpool Overhead Railway. This work is unique in England, though on lines similar to the elevated lines in New York City. Sir Douglas and his brother were responsible for the engineering work on the Southern and Metropolitan divisions of the Great Central Railway. He also took a prominent part in underground-railway development in England. In Africa his name is associated with the Rhodesian Railway and the famous bridge over the Zambesi River at Victoria Falls. Other undertakings in South Africa were the Beira Port and Railway, Benguela Railway and the Trans-Zambesi Railway. In South America he was responsible for the construction of railways in the Argentine for the Central Argentine Railway Co.; in Colombia, for the Dorada Railway Co., and also in Brazil.

He was an enthusiastic advocate of standardization and was one of the early supporters of the movement which resulted in the formation of the British

Engineering Standards Committee in 1901 on whose main committee he served from the time of its formation till early in 1920.

Sir Douglas Fox became a member of the Institution of Civil Engineers in 1866, was elected to the council in 1884 and served as its president in 1899–1900. During his term of office he received the American civil and mechanical engineers, who were entertained at the Guildhall on the occasion of their visit to London that year. He was also a member of the Institution of Mechanical Engineers, the Institution of Electrical Engineers, and an honorary member not only of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS but also of the American Society of Civil Engineers. He was a director of some ten companies, was vice-chairman of the South Indian Railway Co., and chairman of the Industrial Dwellings Co. and of the Northfleet Coal and Ballast Co. He patented numerous inventions connected with railway work. In 1880 he was appointed lecturer to the School of Military Engineering, Chatham, his lectures dealing with light and temporary railways. He was awarded at various times by the Institution of Civil Engineers the Manby Premium, the Telford Medal, and Telford Premium for contributions to the proceedings. He was elected a Fellow of King's College (London) in 1887. He was a justice of the peace and took a prominent part in many of the religious and philanthropic movements of the day.

FREDERICK FRELINGHUYSEN GAINES

Frederick F. Gaines, a member of the Railroad Administration Labor Board until its disbandment in 1920, died at his home in Washington, D. C., on August 26, 1921. Mr. Gaines was born at Hawley, Pa., on March 28, 1871. He was prepared for college at Wyoming Seminary and entered Cornell University in 1891, receiving his M.E. degree in 1895. He specialized in locomotive engineering and during his summer vacations worked in the locomotive repair shops of the Erie & Wyoming Railroad at Dunmore, Pa.

Thus began his active connection with railroading which continued until his death. Upon graduation he entered the employ of the Lehigh Valley Railroad, being successively draftsman, mechanical engineer and master mechanic. After serving with the Philadelphia and Reading as master mechanic at Reading, he became superintendent of motive power of the Central of Georgia Railroad at Savannah, Ga., which position he held until the formation of the U. S. Railroad Administration when he was appointed mechanical superintendent of the Southern Region with headquarters at Atlanta. Upon the creation of the Railroad Administration Labor Board, he was made a member and served until that body was disbanded in 1920.

Mr. Gaines was a member of the Railway Master Mechanics Association and the Master Car Builders' Association of both of which he was past president. He was a Shriner, a Knight Templar, a member of the Society of Colonial Wars and of the Sons of the Revolution. He became a member of our Society in 1901.

ALONZO GARTLEY

Alonzo Gartley, vice-president, manager and director of C. Brewer & Co., Honolulu, Hawaii, and one of the foremost men in Honolulu's industrial and financial life, died on April 21, 1921. Mr. Gartley was born on October 14, 1869,

in Cedar Falls, Iowa. He was educated in the U. S. Naval Academy at Annapolis, being graduated in 1890 as an ensign. He resigned from the Navy two years later and became associated with the Cramp Shipyard, Philadelphia, Pa.

During the Spanish War he was commissioned a lieutenant in the Navy and commanded the converted yachts U.S.S. *Aileen* and *Sylph*. He was also in command of the Pennsylvania Naval Reserve from 1897 to 1900. After the war he took the position of gas engineer with the United Gas Improvement Co., Philadelphia. He was next associated with the Hawaiian Electric Co., Ltd., as manager, which position he held until 1910 when he became consulting engineer for C. Brewer & Co., later becoming vice-president, manager and director of the firm which is the oldest business house of the Hawaiian Islands.

In 1908 Mr. Gartley was appointed a member of the first conservation commission which was sent from Hawaii to Washington. He was actively interested in the public affairs of Hawaii and filled a number of public positions very ably. He was a member of the University, Commercial, Country and Pacific Clubs. He belonged to the American Institute of Electrical Engineers, the American Gas Institute and the Honolulu Engineering Association. He became a member of our Society in 1911.

GEORGE NATHAN GIFFORD

George N. Gifford, assistant plant engineer, the American Printing Co., Fall River, Mass., died on May 1, 1921. Mr. Gifford was born on July 25, 1887, in Dartmouth, Mass. He attended the Fall River schools and later entered Worcester Polytechnic Institute from which he was graduated in 1910 with the degree of B.S. Upon graduation he was for one year with the Sullivan Machinery Co., Claremont, N. H., as salesman of quarrying and mining machinery, then becoming associated with the American Printing Co., as assistant to the plant engineer, which position he held at the time of his death.

Mr. Gifford became a junior member of the Society in 1913 and in 1918 was advanced to the grade of associate member. He was a member of several fraternal organizations.

EMILIO S. GODOY

Emilio S. Godoy, vice-president and general manager of the Standard Shipbuilding Corporation, Shooter's Island, N.Y., died on January 12, 1921. Mr. Godoy was born on October 15, 1872, in Santiago, Cuba. He was graduated from the Institute of Havana in 1889 and two years later entered the street-railway field as manager of the city system of Lima, Peru.

He projected the first electric railway in Peru, a twenty-mile interurban line from Lima to Chorrillos and managed its construction in 1904. He was one of the organizers of the Empresas Electricas Asociadas of Lima, founded in 1906. He served on the board of managers until 1913 when he accepted the appointment of general manager of the Santiago system. While in Santiago Mr. Godoy was also manager and director of the Compania de Urbanizacion y Ensanche de Santiago y Caney, a land development company, responsible for the development and construction of the most modern part of the city. He held both positions until the latter part of 1916 when he was appointed general manager and vice-president of the Standard Shipbuilding Corporation, Shooter's

Island, Staten Island, N.Y., which he reorganized, increasing its personnel during the war to 7,000 men and launching 27 ships.

Mr. Godoy became a member of the Society in 1919.

WEBSTER V. GOULD

Webster V. Gould, a member of the Society since 1887, died on December 1, 1921. Mr. Gould was born in Norwich, Conn., on May 31, 1854, and was educated in the Norwich schools, later attending the Bryant and Stratton School of Mechanical Engineering in Providence, R. I. He served his apprenticeship with C. B. Roger and Co., and upon its completion became correspondence clerk and draftsman and then superintendent of the company. In 1892 he became associated with the Jones and Lamson Machine Co., Springfield, Vt., remaining as their representative for twenty years, retiring in 1912. In 1914, upon moving to New Haven, he entered the employ of the Winchester Repeating Arms Co., serving in the purchasing and scheduling departments until the time of his death.

Mr. Gould became a member of the Society in 1887. He was also a member of the Machinery Club of New York.

WILLIAM E. HASWELL

William E. Haswell, superintendent of Richards, McCarty and Bulford, Columbus, Ohio, died in November 1921. Mr. Haswell was born in June 1871, in Circleville, Ohio. He was educated in Wittenberg College.

His first employment was on engineering construction in the Circleville office of S. R. Bullock and Co., Corry, Pa. He was for a short period with the Holly Manufacturing Co., Anniston, Ala., then becoming connected with the Circleville Water Works as chief engineer and assistant superintendent. From 1902 to 1908 he held the position of State Examiner of Engineers for the eighth district of Ohio with headquarters in Bucyrus, resigning to become secretary and manager of the Hall Gas Engine Co. For two years Mr. Haswell was chief examiner of engineers in Columbus, Ohio, when he became secretary of the Ohio Board of Administration which position he held from 1911 to 1914. In 1914 he accepted the position of superintendent of Richards, McCarty and Bulford where he was located at the time of his death.

Mr. Haswell became a member of the Society in 1914.

GEORGE R. HENDERSON

George R. Henderson, long prominent in the engineering profession and a member of the Society since 1890, died on October 19, 1921. Mr. Henderson was born on January 14, 1861, in Philadelphia, Pa., and was educated in the schools of that city. He began his career in the Pennsylvania Railroad shops at West Altoona and then was associated with the Norfolk & Western and Santa Fé Railroads, the Schenectady Locomotive Works and the Baldwin Locomotive Works.

Later he became consulting engineer for railroads in Brazil, the Philippine Islands and for several roads in the United States. In 1918 he was appointed Federal Fuel Administrative Engineer for Pennsylvania. He was the author of several engineering textbooks and other technical works.

Mr. Henderson became a member of our Society in 1890. He was also a member of the Franklin Institute and one of its managers, serving since 1912 on its Committee on Science and the Arts, being chairman in 1914–15.

Resolutions were passed by this Committee in which it is stated that the Committee "lost one of its most valued members; one who could always be depended upon to act intelligently, willingly, indefatigably and with a clear vision in all matters referred to him by this Committee. . . . His wide experience, broad knowledge, sound judgment and willingness to help made him a valued member, and his death is a serious loss to this Committee."

JACK STANLEY HERBERT

Jack S. Herbert, assistant superintendent of the Gautier Department of Cambria Steel Co., Johnstown, Pa., died on February 24, 1921. Mr. Herbert was born in Newcastle, Wyo., on March 13, 1881. He was educated at West Point Military Academy and Massachusetts Institute of Technology. He served his apprenticeship with the Central of Georgia Railway from 1895 to 1898, then being employed as draftsman for the Consumers' Electric Co., Tampa, Fla.

For one year he was with the William R. Kiddell Co., Sioux City, Iowa, as assistant to the chief engineer and then became engineer of construction for the Buckhorn Portland Cement Co. In 1905 he took the position of engineer of maintenance and construction for the Pennsylvania Portland Cement Co. Later he was president and general manager of the Herbert-Evans Co., in Wilkinsburg, Pa. In 1913 he was appointed superintendent of safety for the Cambria Steel Co. and in 1917 became assistant to the head of the Gautier Department which position he was holding at the time of his death.

Mr. Herbert served in the Rough Riders under Colonel Roosevelt during the Spanish-American War. He was a member of the Aero Club of America, holding a pilot's license. He also belonged to the Circumnavigators' Club of New York. He became a member of the Society in 1908.

FREDERICK W. HOLLMAN

Frederick W. Hollman, master mechanic, Bethlehem Steel Co., Sparrows Point, Md., died in January 1921. Mr. Hollman was born in New York City in July 1863. He was educated in the public schools of the city and Columbia University from which he received his M.E. degree in 1905. In the fall of that year he became associated with the Maryland Steel Co., Sparrows Point, Md., where he remained until 1909 when he became assistant mechanical engineer of the Carborundum Co., Niagara Falls, N. Y.; shortly afterwards he was advanced to the position of works engineer and master mechanic. In 1913 Mr. Hollman returned to the Maryland Steel Co. as master mechanic and three years later became associated, in the same capacity, with the Bethlehem Steel Co.

Mr. Hollman became a junior member of the Society in 1907 and attained full membership in 1914.

W. E. HOLMES

Winfield E. Holmes, who became a member of THE, AMERICAN SOCIETY OF MECHANICAL ENGINEERS in 1914, died on December 25 1921, at his home in

Springfield, Mass., following a five months' illness. Mr. Holmes was born in Yarmouthport, Mass., on May 10, 1881. He was a graduate of Worcester Polytechnic Institute, class of 1904, and had been connected with the S. M. Green Co. since his graduation, being rated as one of the city's foremost engineers and an authority on industrial processes, methods and equipments.

Mr. Holmes was active in many Springfield clubs and fraternal organizations; he was vice-president of the Rotary Club and was one of the group responsible for the organization in 1919 of the Engineering Society of Western Massachusetts, holding the office of secretary-treasurer since that time. He was also a member of the Engineers' Club of New York.

RICHARD HUTCHISON

Richard Hutchison, district sales manager of the Boston office of Babcock & Wilcox Co., died on June 15, 1921. Mr. Hutchison was born on May 25, 1872, in Newcastle, N. B., Canada. He was educated in the schools of Annapolis Valley, N. S., attending for a short period Acadia College, and later the Rhode Island Technical Drawing School. In 1894 he became associated with the Babcock & Wilcox Co. as draftsman in their New York office; two years later he was made chief draftsman. In 1898 he was transferred to the Boston office of the company and appointed district sales manager which position he held at the time of his death.

Mr. Hutchison was a member of the Boston Society of Civil Engineers, the Engineers' Club of Boston and of a number of other similar organizations. He also belonged to the Masonic fraternity. He became an associate member of the Society in 1900 and was promoted to the grade of member in 1904.

CHARLES MAPLES JARVIS

Charles Maples Jarvis, for some years president of the American Hardware Corporation, New Britain, Conn., and for twenty-two years before that prominently identified with fabricated steel work, died on May 21, 1921, after an illness of two years. Mr. Jarvis was born on April 16, 1856, at Deposit, Delaware County, N. Y. He was graduated from the Sheffield Scientific School of Yale University in 1877 as a civil engineer. He then entered the employ of the Corrugated Metal Co., East Berlin, Conn., as engineer and draftsman. Later he became chief engineer and president of the company and on its absorption by the American Bridge Co. became a director of the latter as well as a vice-president and member of the executive committee. He retired when the company became a subsidiary of the U. S. Steel Corporation. Mr. Jarvis became a member of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS in 1890 and served as vice-president from 1897 to 1899.

GROVER R. JASPER

Grover R. Jasper, connected with the firm of Smith, Ames & Chisholm, consulting engineers, Havana, Cuba, died on May 10, 1921. Mr. Jasper was born in Indianapolis, Ind., on December 18, 1885, and was educated in the schools of that city. His first work was with the Link Belt Co., Indianapolis, as tracer,

later becoming a draftsman. In 1907 he was engaged by the Nordyke & Marmon Co. as tool designer, returning shortly to the Link Belt Co. as draftsman in charge of their tool-designing department. In 1910 Mr. Jasper accepted the position of draftsman on work in connection with the Panama Canal and in 1913 was made assistant chief draftsman of the mechanical division and later chief planner of the division.

In 1917 Mr. Jasper became purchasing inspector with the Emergency Fleet Corporation, Washington, D. C., where he was located until early in 1921 when he resigned to accept a position with the firm of Smith, Ames & Chisholm.

Mr. Jasper became an associate member of the Society in 1918.

SAMUEL SPRING JORDAN

Samuel S. Jordan, naval architect and engineer, and a member of our Society since 1887, died on January 5, 1921. Mr. Jordan was born in Cape Elizabeth, Me., on June 26, 1862. He was a graduate in mechanical engineering of Worcester Polytechnic Institute. He spent about a year in pattern making with the Star Match Co. and the Portland Co., Portland, Me., and another year setting up marine engines in the works of the latter company. He returned to the Star Match Company where for two and a half years he designed, built and ran special machinery. During part of that time he had supervision of the machine shop and automatic machinery.

In March 1886, Mr. Jordan entered the employ of John Roach of Philadelphia in charge of the engine work and drafting in his shipyard. The following year he accepted a position as first-class draftsman to work on designs of triple-expansion engines for cruisers 4 and 5. These engines were built at the Union Iron Works in San Francisco, Cal. Later Mr. Jordan was assistant superintendent in the shipyard at Newport News, Va., and during his stay there the cruisers *New York* and *Brooklyn* were built. He resigned from this position to become superintendent of the Quintard Iron Works, New York, N. Y., where he was located until 1903 when he opened offices as a consulting engineer and naval architect. For the past fifteen years Mr. Jordan was managing the family estate in Scarboro, Me.

JAMES KEITH

James Keith, managing director of James Keith & Blackman Co., Ltd., London, England, died on February 23, 1921. Mr. Keith was born in Arbroath, Scotland, in February, 1849. He was educated in the Arbroath schools and then served an apprenticeship in his father's works. Upon the completion of this training he visited Canada and the United States to gain further engineering experience. In 1868 he established his own business, specializing in heating, ventilating, gas and hydraulic engineering. In 1900 his business was amalgamated with that of the Blackman Ventilating Co., Ltd., and he became managing director of the combined concerns, the successful career of which under his direction is well known.

Mr. Keith was the pioneer in Scotland in the engineering trade of the fifty-one hour and later the forty-eight hour working week. He was the inventor of numerous hydraulic, heating, ventilating and other appliances, among the

earliest of which was the apparatus for the successful and economical manufacture of mineral-oil gas.

Mr. Keith was an associate member of the Institution of Civil Engineers, a member of the Institution of Mechanical Engineers, the Institution of Marine Engineers, the Royal Institution of Great Britain and the Institute of Industry of Great Britain. He became a member of our Society in 1916.

LUCIUS J. KNOWLES

Lucius J. Knowles, president of the Crompton & Knowles Loom Works, Worcester, Mass., died suddenly on October 26, 1920, in London, England. Mr. Knowles was born in Worcester, Mass., on April 6, 1879, and received his early education in the schools of that city. He was prepared for college at Dalzell Preparatory School and Worcester Academy and then entered Harvard University, leaving at the close of his junior year.

In 1903 he became associated with the business which his father and uncle had established in 1866, the Crompton & Knowles Loom Works. From 1906 to 1917 he was treasurer and vice-president of the company and at the end of that period succeeded to the presidency of the firm. Mr. Knowles was a director in several concerns and was prominent in the club life of Worcester and surrounding cities. He became an associate of the Society in 1918.

CHARLES G. LEESON

Charles G. Leeson, construction engineer for the new Hutchinson Lumber Co. plant at Oroville, Cal., died on November 25, 1921. Mr. Leeson was born on February 14, 1874, in Manchester, Mich., where he received his early education. Later he attended the University of Michigan, receiving his B.S. in mechanical engineering in 1901.

His first employment was as draftsman and erecting engineer with the Underfeed Stoker Co. of Chicago where he was located until 1903 when he became connected with the Bucyrus Co., Milwaukee, as draftsman; in 1904 he was advanced to the head of the production department and two years later to the repair department. In 1907 he accepted the position of shop superintendent with the Folsom Development Co. from which he resigned in 1909 to become assistant manager of the dredging department of the Oro Water, Light and Power Co.; in 1912 he was appointed manager of the department. Five years later this concern became the American Gold Dredging Co. and Mr. Leeson was made operating manager in full charge of their Shasta, Calaveras and Butte County gold dredges, which position he occupied until 1920 when he became engineer and chief draftsman for the Schmeiser Manufacturing Co., Davis, Cal. At the time of his death he was engaged in construction work in Oroville, Cal.

Mr. Leeson became a member of the Society in 1913. He was a former member of the Union League Club of San Francisco. For two years he was president of the Oroville Chamber of Commerce; he was chairman of the Lincoln Highway Association of Butte County, Cal.; he was on the Board of Directors of Pike's Peak Ocean to Ocean Highway and was interested in a number of other civic organizations. He belonged to the Masonic Order, was a Knight Templar and a Shriner.

WILLIAM WILSON LIGHTHIPE

William W. Lighthipe, engineer for the Otis Elevator Company and well-known as an elevator expert, died August 7, 1921, in the Long Island Hospital. Mr. Lighthipe was born in 1875 in Vincentown, N. J. He attended Trinity School, New York, and then the School of Mines, Columbia University, from which he was graduated in 1898 with the degree of electrical engineer.

Mr. Lighthipe was employed successively by the Metropolitan Electric Construction Company and the Sprague Elevator Company until 1901, when he became assistant superintendent of the Otis Elevator Company. With the exception of two years, 1903–1905, when he served as superintendent of construction and engineer for the Marine Engine & Machine Co., he was connected with this firm up to the time of his death, first as sales engineer and later as general service manager.

For many years Mr. Lighthipe was an authority on the subject of elevator service in various classes of buildings, and has written a number of valuable papers on this subject, in addition to his many lectures on elevators before noted engineering and educational societies. He became a member of the Society in 1907 and belonged also to the American Institute of Electrical Engineers and many other social and fraternal organizations.

JACOB BOWEN MCKENNAN

Jacob B. McKennan, vice-president and general manager of the Colorado Fuel & Iron Co., Denver, Col., died on March 7, 1921. Mr. McKennan was born in Pittsburgh, Pa., on January 31, 1871. He served a three-year apprenticeship with the Homestead Steel Works and then became mechanical engineer for Henry Aiken in Pittsburgh. For nine months he was in charge of the erection of universal plate mill and equipment for the Central Iron Works, Harrisburg, Pa.

He entered the service of the Colorado Fuel & Iron Co. as draftsman in 1894 and progressed through various positions, including night foreman of the rail mill and superintendent of the blast-furnace department. He was appointed assistant general superintendent of the steel mills in 1899. Nine months later he was made superintendent and in 1901 assistant manager, becoming manager in 1906. In 1915 Mr. McKennan was appointed general manager of the whole company and in 1917 took the added duties of the vice-presidency.

Mr. McKennan became a junior member of the Society in 1899. He was also a member of a number of clubs in Denver.

HARRY PEAKE MACDONALD

Harry P. Macdonald, vice-president and chief engineer of Snead & Co., Jersey City, N. J., died on August 25, 1921. Mr. Macdonald was born on May 19, 1880, in Louisville, Ky. He attended high school in Newton, Mass., and then entered the Massachusetts Institute of Technology from which he was graduated in 1901 with the degree of B.S. in civil engineering.

He entered the employ of the Snead & Co. Iron Works in 1901 and served consecutively as draftsman, assistant superintendent and superintendent, be-

coming vice-president and chief engineer in 1915. Mr. Macdonald held a number of patents covering the Macdonald roller, ramming and pattern-drawing molding machines for foundries, the electrical heat treating of steel and other metals by means of internal resistance and the use of electricity in the vaporization of gasoline for gas-engine service.

He was a member of the Society of Automotive Engineers, the American Society for Steel Treating, chairman of the safety and sanitation committee of the National Foundries Association, chairman of the membership committee of the Technology Club of New York, a member of the Jersey City Chamber of Commerce, of the Sons of the American Revolution and of the Sons of the Colonial Wars. During the war he served on the Committee for Industrial Training for War Emergency under the Council of National Defense and as adviser to the Metal Trades Board on War Labor Policies. Mr. Macdonald became a member of our Society in 1914.

GUY L. MEAKER

Guy L. Meaker, formerly president of the Meaker Galvanizing Co., Chicago Ill., died on August 7, 1921. Mr. Meaker was born in Detroit, Mich., on September 6, 1873. He was educated in the schools of Wellington, Ohio, and served his apprenticeship with the Meaker Manufacturing Co., Chicago. His further education was obtained through his own efforts, entirely, and he was a constant student along chemical and mechanical lines, taking a correspondence course in Columbia University.

Mr. Meaker's chief work was in connection with the electro-deposition of zinc, lead, etc. He was the founder of the Meaker Galvanizing Co. and was for several years actively engaged with the American Steel & Wire Co. in improving their galvanizing processes. With them he developed an electro-galvanizing equipment which was operated successfully and he was also largely instrumental in improving and standardizing the hot galvanizing methods employed with wire.

During the last eight or ten years of his life his activity was greatly handicapped by the loss of his eyesight but in spite of this he continued his studies and investigations, and, with assistance, developed a considerable practice in consultation work in connection with galvanizing problems.

Mr. Meaker became a junior member of the Society in 1904.

BARCLAY GEORGE MERING

Barclay George Mering, industrial engineer of Indianapolis, Ind., and mechanical engineer of the American Hominy Co. also of that city, died on February 20, 1921. Mr. Mering was born in Cincinnati, Ohio, in July 1864. He was educated in the schools of Terre Haute, Ind., and attended Rose Polytechnic Institute where he received his B.S. degree in 1887; ten years later he received his M.S. from the same institution.

Upon graduation he spent one year with the Deane Steam Pump Works, Holyoke, Mass., and then became connected with the Eagle Iron Works, Buffalo, N. Y., as draftsman. He was with this company for five years when he resigned to become head of the engineering department of the Quaker Oats Co., Chicago,

Ill. Two years later he took charge of the engineering department of the American Hominy Co., maintaining at the same time his own office as consulting engineer in industrial engineering, designing buildings, power plants and machinery layouts for industrial plants, etc.

Mr. Mering was a member of the Chamber of Commerce of Indianapolis. He belonged to the Indianapolis Optimists' and the Hoosier Motor Clubs. He became a member of our Society in 1913.

JAMES I. MILLIKEN

James I. Milliken, agent since 1895 of the Everett Mills, Lawrence, Mass., died on April 19, 1921, at his home in that city. Mr. Milliken was born on March 9, 1850, in Saco, Me. He received his early education in the public schools and Monson Academy, Monson. At the age of twenty-one he was appointed overseer of the finishing department of the Slater Mills, Slatersville, R. I., the oldest cotton mills in the country. His next position was that of overseer of finishing from which he was advanced to the assistant superintendency at the Palmer Mills in Three Rivers, Mass. In 1888 Mr. Milliken accepted the position of superintendent of the Everett Mills and in 1895 he was appointed agent, holding that position until the time of his death.

Mr. Milliken was a practical manufacturer of marked ability and was especially successful in dealing with the human element among his mill employees. He was very much interested in church work and charitable organizations and gave much of his time and thought to that work. He belonged to a number of fraternal and social organizations and clubs, including the Masonic fraternity, the Chamber of Commerce and the National Association of Cotton Manufacturers. He became a member of our Society in 1917.

WILLIAM DOUGLAS MONKS

William D. Monks, president of the Hudson Ice Co., Jersey City, N. J., died on December 28, 1921. Mr. Monks was born in New York City on February 18, 1886, and was educated in the city schools, later attending Columbia University from which he was graduated as a mechanical engineer in 1907. A year later he received his M.A. degree.

He was first employed by the De La Vergne Machine Co., becoming in 1910 assistant chief engineer of the company; in 1914 he was made sales engineer, which position he held until 1921 when he resigned to become associated with the Hudson Ice Co.

Mr. Monks was a member of the American Society of Refrigerating Engineers and of several clubs. He became a junior member of our Society in 1908.

STANLEY H. MOORE

Stanley H. Moore, formerly director of the mechanical arts department, McKinley High School, St. Louis, Mo., died in Los Angeles, Cal., on January 15, 1921. Mr. Moore was born in St. Louis, Mo., on November 29, 1874. He received his early education in the public schools of St. Louis and his technical training in Washington University of that city. Having obtained shop ex-

perience with the Rankin Fritsch Co., St. Louis, and the Webster, Camp & Lane Co., Akron, Ohio, he became director of manual training in the Manual Training High School of Kansas City, Mo. Later he was associated in the same capacity with the McKinley High School of St. Louis where he remained until ill health forced him to resign.

Mr. Moore was the author of many technical papers and of a book on Mechanical Engineering and Machine-Shop Practice which is widely used not only in the shops but in a number of universities as a textbook.

He became a junior member of the Society in 1900 and an associate in 1903. He was also a member of the American Institute of Electrical Engineers, the Society for the Promotion of Engineering Education and the National Educational Association.

JOHN MULLEN

John Mullen, president of the National Bank of Shamokin, Pa., since 1889, and former head of John Mullen & Son, iron workers, foundrymen and machinists, died February 22, 1921, after a brief illness. Mr. Mullen was born February 27, 1838, at Port Carbon, Pa. At the age of ten he began an apprenticeship as a machinist with T. H. Winterstein at Port Carbon, with whom he remained for fifteen years, becoming a master machinist and patternmaker.

He then became a member of the firm of Robert Allison & Co., iron molders and machinery manufacturers of Port Carbon, which shortly afterward lost its entire plant by fire. When the shops were rebuilt Mr. Mullen was appointed foreman and remained with the firm in that capacity until 1870 when he removed to Shamokin and became a member of the firm of Mullen and Huffman, who leased and conducted the Shamokin Iron Works. The firm was later known as Mullen and Bittenbender, John Mullen and Co., and finally as John Mullen and Son, and the Mullen Foundry was considered to be one of the most modern of its time.

Mr. Mullen was associated with Thomas Edison in the erection of the first completed electrical plant in the United States and was treasurer of this corporation, the Edison Electric Light Co., until it was absorbed by what is now the Pennsylvania Power and Light Co. He was also one of those who organized the Shamokin Coal and Coke Co. and revolutionized the mining and preparation of soft coal.

The Shamokin Powder Company was also organized by Mr. Mullen and he was interested in The Shamokin Steam Heating Co., the Shamokin Mfg. Co., the Majestic Coal Co., and the Thomas Devlin Mfg. Co., of Philadelphia.

He served in the Ninth Regiment, Pennsylvania Volunteers, during the Civil War and throughout his life was especially active as an official in the work of the G.A.R. He also devoted much time to civic affairs, having a keen interest in all matters pertaining to public service. He became a member of the Society in 1886.

KUMATARO NAKAI

Kumataro Nakai, engineer and chief of department of the Ashio Copper Mines, Furukawa & Co., Ltd., Tokyo, Japan, died in the latter part of 1921. Mr. Nakai was born in June 1876, in Matsuye, Shimaneken, Japan, and was graduated as a mechanical engineer from the Kioto Imperial University. In

1901 he was appointed lecturer in the University where he was located until 1903 when he became associated with Furukawa & Co. During the winter of 1909–1910 Mr. Nakai made an extended tour of investigation for this firm through Germany, Austria, Switzerland, Belgium, England and the United States to study the mining machinery and methods of these countries.

Mr. Nakai became a member of the Society in 1914.

LEWIS W. NEWTON

Lewis W. Newton, consulting engineer, Enameled Metals Co., Etna, Pa., died on January 12, 1921. Mr. Newton was born in Sterling, Mass., on March 23, 1854. He received his early education in the grammar schools of Sterling and in Monson Academy and later attended Tufts College. He left Tufts to enter Amherst where he spent several terms when he left to become a partner in his father's business of tanning.

In 1878 he took up the machinist trade and served his apprenticeship with the Clinton Wire Cloth Co., studying at night to become a mechanical engineer. He served as foreman and superintendent in several machine shops, among them the C. W. C. Co., the Holyoke Machine Co., the Simonds Rolling Machine Co. and the Ferracute Machine Co., where he designed and installed their large die presses. For eight years he was connected with Charles T. Schoen of the Pressed Steel Car Co., as general superintendent of shops. In 1893 he became associated with the Standard Horse Nail Co., designing and installing the automatic mill-feeding machines. Ten years later he opened consulting offices in Pittsburgh, Pa. In December, 1919, he became consulting engineer for the Enameled Metals Co., Etna, Pa.

Mr. Newton became a member of the Society in 1903.

WILLIAM EDWARD NINDE

William E. Ninde, associate professor of mechanical engineering in the College of Applied Science, Syracuse University, died on February 20, 1921. Professor Ninde was born in Rome, N. Y., on December 3, 1865; he attended the city schools including the Rome Free Academy. In 1885 he entered the drafting department of the New York Locomotive Works in Rome where he served his full apprenticeship and remained for some time afterwards. For two years he was connected with the American Locomotive Co., Schenectady, as a designer, then taking a position in the same capacity with McIntosh & Seymour Co., Auburn, N. Y.

After three years with the Compressed Air Co. of New York, he became, in 1901, assistant engineer and chief draftsman of the Bass Foundry & Machine Co., Fort Wayne, Ind., remaining four years. He then entered the engineering department of the Farrell Foundry & Machine Co., Ansonia, Conn., resigning to accept a position with the Solvay Process Co., Syracuse. In 1908 he became an instructor in the L. C. Smith College of Applied Science, Syracuse University, and was soon advanced to the position of associate professor of mechanical engineering.

Except for a two-year course in the Y.M.C.A. College, Springfield, Mass., which was non-engineering, and a course in experimental engineering in Cornell

University during the summer of 1898, Professor Ninde was a self-educated man. By his persistency and application to study he succeeded in his chosen profession. In 1914 he received from Syracuse University the honorary degree of mechanical engineer. He was elected an honorary member of Tau Beta Pi. For a number of years he served as secretary of the faculty of the College of Applied Science. He was the author of Design and Construction of Heat Engines published by the McGraw-Hill Book Co.

In a tribute to his memory Dean Graham of the College of Applied Science, says:

> "Professor Ninde had many rare qualities and the combination made him a most delightful associate and friend. One of his most striking characteristics was his entire self-forgetfulness. No task was too difficult for him to undertake, no burden too heavy for him to assume. All who knew him felt that he made the Golden Rule his rule of life — not in any ostentatious way, but perhaps because he could not help it."

Professor Ninde became a junior member of the Society in 1899 and was advanced to full membership in 1903. He was instrumental in organizing and directing THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS Student Branch of Syracuse University and was its honorary chairman. He was vice-chairman of the Syracuse Section of the Society.

GEORGE A. O'ROURKE

George A. O'Rourke, chief engineer, N. Y. State Industrial Commission, Albany, N. Y., died on April 24, 1921. Mr. O'Rourke was born in New York City on July 7, 1870. He was educated in the public schools of the city, Cooper Union and the College of the City of New York of which he was a graduate. For twenty years Mr. O'Rourke was associated with the Weil & Mayer Construction Co., New York City, where as general superintendent he was in charge of construction of many large buildings, including the installation of steam plants.

For a time he was with the Rapid Transit Commission of New York City and for three years following the earthquake in San Francisco was managing head of the Thompson-Starrett Co. in charge of reconstruction work in that city. For five years Mr. O'Rourke was in the contracting business for himself in New York City, later becoming construction superintendent for the O'Rourke Engineering Co.

In the fall of 1914 when New York State established the office of State Fire Marshal Mr. O'Rourke was appointed chief engineer of that department and continued in that capacity until April of the following year when the department was legislated out of office. Many of the duties of the department, including the inspection of boilers and the licensing of powder magazines, were transferred at that time to the State Department of Labor through the creation of a Board of Boilers and Explosives. In June 1915, when the new bureau began functioning Mr. O'Rourke was appointed as chief engineer, continuing as the head of that Bureau until the time of his death.

During the war, the Federal Government asked for his services and he was given a leave of absence for duty in the Ordnance Department of the U. S. Army, having the rank of major. In that capacity he rendered most valuable service.

Mr. O'Rourke became a member of the Society in 1918.

FRANCIS W. ORPIN

Francis W. Orpin, engaged in engineering work in Montevideo, Uruguay, died in 1921. Mr. Orpin was born in San Diego, Cal., on July 10, 1891. He was educated in San Francisco schools and attended the School of Industrial Arts in that city. He was employed on various engineering construction works in California and Oregon, then becoming third assistant engineer on the S.S. *Uganda*, under the management of McLay and McIntyre, Glasgow, Scotland.

From 1912 to 1919 Mr. Orpin held the following positions: engineer, Terminal Railroad and Warehouse Co., Chicago, Ill.; oiler and engineer, Portland Railroad Light and Power Co., Portland, Ore.; engineer on the operation of steam and turbo-electric machinery, Portland Gas and Coke Co.; machinist on construction and maintenance, plant of the Columbia River Shipbuilding Corporation, Portland, Ore.; chief machinist and mate of the U.S.S. *South Dakota*, having charge of upkeep of main engines; assistant engineer, U.S.S. *South Dakota*, in charge of the boiler-room station; U. S. Navy, commissioned as ensign in the U. S. Naval Reserve Force and appointed as chief engineer on the U.S.S. *Helena*. In 1920 Mr. Orpin became interested in engineering work in Montevideo, Uruguay, in which work he was engaged at the time of his death.

Mr. Orpin became an associate member of the Society in 1920. He was also a member of the National Association of Stationary Engineers.

WILLIAM ROBERT PARK

William Robert Park, consulting engineer for the Hancock Inspirator Company of Boston, Mass., died in Taunton, Mass., on July 28, 1921. Mr. Park was born in Brooklyn, Conn., August 29, 1831. He became an orphan at the age of ten and at sixteen left school to enter a machine shop at North Scituate, Mass., where he served a three-year apprenticeship. He then entered the employ of the Taunton Locomotive Mfg. Co. and later the Mason Machine Works, where he remained until 1857.

About this time Mr. Park became interested in the manufacture of sewing machines. In 1858 he went to Boston to the Grover and Baker Sewing Machine Co., with which he remained until 1876, when the firm dissolved. From then up to the time of his death Mr. Park was connected with the Hancock Inspirator Company, serving as superintendent and manager and finally as consulting engineer. He also founded the firm of W. R. Park & Son, plumbers and steamfitters, of Taunton.

Mr. Park's inventions were so numerous and valuable that he was awarded a gold medal by the United States government for his achievements in mechanics. He was a life member of the Society, which he joined in 1889.

CECIL P. POOLE

Cecil P. Poole, city mechanical engineer of Atlanta, Ga., died on February 23, 1921. Mr. Poole was born in Elizabeth City, N. C., in October 1865. He received his early education in southern schools and then, being gifted with a strong talent for mathematics, determined to educate himself for a technical career.

He worked at night as a telegraph operator and at the same time pursued his studies along technical lines. At the completion of his course he was given a position with the Western Union Telegraph Co. and within a year he became manager and chief engineer of the electric lighting company in Lynchburg which position he held for ten years.

During this period his articles for engineering periodicals attracted much attention with the result that he was offered and accepted the position of associate editor of *Electrical World* in New York in 1895. In 1900 he was made editor of the *American Electrician* and in 1905 became editor-in-chief of *Power*.

He removed to Atlanta in 1912 to improve his health and opened an office in that city as consulting engineer. Within a short period he was tendered the position of city engineer where his work proved of very great value and his laboratory a distinct adjunct to all departments.

Mr. Poole was a Fellow of the American Institute of Electrical Engineers, a member of the Society of Automotive Engineers and also of the American Society for Testing Materials. He became a member of our Society in 1909.

HENRY C. PROCHAZKA

Henry C. Prochazka, mechanical engineer with the Universal Oil Products Co., Wood River, Ill., was instantly killed on December 5, 1921, when the plant in which he was working was wrecked by an explosion. Mr. Prochazka was born in Manitowoc, Wis., on August 29, 1890, and attended the schools of that city. Later he was graduated as a mechanical engineer from the University of Wisconsin, class of 1912.

His shop experience was acquired at the Allis-Chalmers plant at West Allis, Wis. For a few months he was with the Wells Power Co., Milwaukee, resigning to become assistant engineer at the heating plant of the University of Wisconsin. During the war he was in the Chemical Warfare Division of the Army, stationed at Willoughby, Ohio, and upon his discharge became associated with the Milwaukee Electric Railway & Light Co., engaged in the planning and construction of the Lakeside Power Plant. When this work was completed he accepted the position of mechanical engineer with the Universal Oil Products Co. who were installing a plant for the Roxana Mining Co. at Alton, Ill., where the accident occurred.

Mr. Prochazka became an associate member of the Society in 1921.

JOSEPH H. ROACH

Joseph H. Roach, president of Joseph H. Roach & Co., now the Roach Stoker Co., Philadelphia, Pa., died on January 11, 1921. Mr. Roach was born on January 18, 1873, in Philadelphia. He was educated in the Friends Central High School of that city and was then employed by the Philadelphia Carey Manufacturing Co. as sales engineer. In 1906 he became connected in the same capacity with W. K. Mitchell & Co. where he was located for three years when he became district sales representative and branch manager for the Hooven, Owens, Rentschler Co.

He resigned from this position in 1916 to organize the concern then known as Joseph H. Roach & Co. and now as the Roach Stoker Co., to manufacture

and sell underfeed standard and simplex stokers which he had designed and invented.

Mr. Roach was a member of the Union League, Manufacturers' and Engineers' Clubs, all of Philadelphia. He became an associate of the Society in 1919.

AUBREY G. ROBB

Aubrey G. Robb, chief engineer and superintendent of the Robb Engineering Works, Amherst, N. S., Canada, died suddenly on June 20, 1921, from blood poisoning. Mr. Robb was born on October 4, 1870, in Amherst, N. S., Canada. He was educated in the Amherst schools and attended the Massachusetts Institute of Technology for one year. In 1891 he entered the Robb Engineering Works, first working on engine design and later becoming chief engineer and superintendent which positions he held at the time of his death.

Mr. Robb served as mayor of Amherst for the year 1909. He also served on the Town Council and the Water Committee. For a number of years he was secretary of the maritime branch of the Canadian Manufacturers' Association. He became a member of the Society in 1902.

HUGO BERNARD ROELKER

Hugo B. Roelker, who died in Port Orange, Fla., on April 18, 1921, was born in Osnabruck, Germany, on September 19, 1843. After obtaining his early education in a technical school, he served his apprenticeship in a machine shop and later worked in the employ of an instrument maker. He then became employed as mechanical draftsman in an engineer's and architect's office. At the age of eighteen he came to this country and after a year's experience with the language and method of working, secured a position as draftsman with the DeLamater Iron Works, New York City. He rose rapidly, becoming chief draftsman, assistant superintendent and, in 1883, superintendent.

Mr. Roelker enjoyed the acquaintance and confidence of many of the leading engineers of the country in developing and supervising the work which they had done at the DeLamater Iron Works. He saw several vessels of the Monitor type built there, including the largest of them, the *Dictator*. These works also fathered the screw propeller, which Captain John Ericcson, the technical genius of the firm, brought to this country. Many of the large industries of the day started in Mr. Roelker's office, — sugar mills, air compressors, ice machines, etc. The first self-propelled torpedoes, torpedo boats, destroyers and submarine boats were all developed under Mr. Roelker's supervision.

When the works closed down on the death of Mr. DeLamater and Captain Ericcson, Mr. Roelker opened an office in Maiden Lane, doing a general mechanical and marine-engineering practice, and making in later years a specialty of the Allen Dense Air Ice Machine, principally for the U. S. Navy.

Mr. Roelker belonged to the Arion, Liederkranz, Columbia Yacht and Engineers' Clubs. He became a member of the Society in 1889.

KENNETH RUSHTON

Kenneth Rushton, vice-president in charge of engineering, The Baldwin Locomotive Works, died September 2, 1921, at the age of sixty years. Mr. Rushton was born in Philadelphia, Pa., and was educated in the city schools and Episcopal Academy. He served an apprenticeship as machinist under Hugo Bilgram, of Philadelphia, and afterward entered the employ of The Baldwin Locomotive Works, in April 1881.

Mr. Rushton's association with The Baldwin Locomotive Works continued uninterruptedly until the time of his death; first as a draftsman, and later as designer, chief mechanical engineer, and finally as vice-president. He was the inventor of many appliances used in the construction of locomotives, and was closely associated with S. M. Vauclain in the development of the four-cylinder compound that bears the name of the latter. While Mr. Rushton did not travel extensively in the prosecution of his business, he represented Baldwin's abroad on some important missions. In 1913 he was sent to Chile, visiting various points of railroad interest on the west coast of South America, and in 1918 went to France in connection with the design of railway transport for artillery.

Mr. Rushton became a member of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS in 1918, and served actively on the Sub-Committee of its Boiler Code Committee on Boilers and Locomotives. He was also a member of the American Society for Testing Materials.

WILLIAM H. SAYRE

William H. Sayre, president of the American Abrasive Metals Co., died suddenly at his home in Glen Ridge, N. J., on January 6, 1921, from a severe heart attack. Mr. Sayre was born at Mauch Chunk, Pa., in September 1865. He was graduated from Lehigh University as a mechanical engineer in 1886. Beginning his practical experience in railroad building in the Northwest, in which he became associated with John B. McDonald, Mr. Sayre later became well known as president of the International Contracting Co., executing many important dredging contracts in New York and other harbors and being called upon to act as a consulting engineer on the Cape Cod Canal.

He was among the first to apply electric welding commercially, organizing the Federal and the Anthracite Electric Welding Companies. In the safety engineering field he was known for his development of the American Abrasive Metals Co., of which he was president until the time of his death.

Mr. Sayre was a member of the American Society of Safety Engineers and was instrumental in its successful reorganization. He became a member of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS in 1911 and served on its Boiler Code Committee, in which he was especially interested.

EDWARD F. SCHNUCK

Edward F. Schnuck, who, until his retirement in 1919, was production engineer and treasurer of Jabez Burns & Co., New York, died on October 2, 1921. Mr. Schnuck was born in St. Louis, Mo., on September 7, 1870. It was in 1888 that he first became connected with Burns & Sons, engineers and manu-

facturers, and it was in connection with the erection of a coffee-roasting plant in St. Louis.

At the completion of that work he came east on general millwright work for the same firm. He had no engineering education up to that time but from personal study and courses in night school he acquired the necessary knowledge for the work, even to the laying out of complete plants. He became well known to the large coffee-roasting concerns throughout the country and in 1897 accepted an offer from Arbuckle Brothers to take charge as general superintendent of their large coffee plant in Brooklyn, N. Y., where he was located for ten years. He resumed his connection with Jabez Burns & Sons in 1907 when the firm was incorporated and continued as one of its directors until his retirement in 1919 because of ill health.

Mr. Schnuck was a member of the American Institute of Electrical Engineers. He became a junior member of our Society in 1897 and was promoted to the grade of member in 1901.

GEORGE J. SMITH

George J. Smith, mechanical expert with the Galena-Signal Oil Co., New York City, died on September 2, 1921. Mr. Smith was born on August 5, 1867, in Poughkeepsie, N. Y., and was educated in the schools of that city. From his boyhood he followed the pursuit of practical engineering and in 1895 became chief engineer of the Poughkeepsie City and Wappinger Falls Electric Co. where he was located until 1902 when he became engineer in charge of operation of the 74th Street power station of the Interborough Rapid Transit Co., New York City. Three years later he was placed in charge of the operation and maintenance of the entire plant.

In 1906 he was offered and accepted the position of mechanical expert with the Galena-Signal Oil Co. to care for their interests in the Middle West with a territory extending from Illinois to Montana, there supervising the operation of power houses and rolling equipment of a large number of electric and steam railroads. In 1917 he was called to the New York territory to supervise the lubrication of the large plants in New York, New Jersey and Pennsylvania. He was engaged in this work at the time of his death.

Mr. Smith became a member of the Society in 1921.

JOHN SMART SMITH

John S. Smith, of the Mexican Eagle Oil Co., Ltd., died on December 23, 1921. Mr. Smith was born in Perth, England, on November 24, 1873. Educated partly at Perth, he later attended Robert Gordon's College at Aberdeen. He served a six years' apprenticeship with Clyne Mitchell & Co., Ltd., marine engineers of Aberdeen, then becoming associated with Cayzer Irvine & Co. as a sea-going engineer where he soon achieved his chief engineer's certificate. He then joined the staff of the Harbor Engineer of Aberdeen and was engaged for three years on general construction and maintenance work, which, combined with his previous training, fitted him to accept a position in 1903 with S. Pearson & Co., Ltd., on their staff at Malta; at that time they were constructing the new docks and harbor for the British Admiralty.

Very shortly after this Sir Pearson required Mr. Smith's services in connection with the development of the oil fields and refineries which he had just undertaken and from then until the time of his death, Mr. Smith was associated with the Mexican Eagle Oil Co., Ltd. During his career with the company he acted at one time as refinery manager at Minatitlan and again as general manager at Tampico; he bore a large and responsible part in the construction of the fleet of tank steamers known as the Eagle Oil Transport and was a director of this steamship company. He was also a director of the Whitehall Corporation, Ltd., and of the Thames Land Co., Ltd.

Mr. Smith was a member of the Council of the Institute of Petroleum Technologists, an associate member of the Institute of Mechanical Engineers and a member of the consultative committee on petroleum technology of the Sir John Cass Technical Institute. He became a member of our Society in 1912.

JAMES PRENTICE SNEDDON

James Prentice Sneddon, general superintendent of The Babcock & Wilcox Co., Bayonne, N. J., died at the Johns Hopkins Hospital, Baltimore, on June 11, 1921. Mr. Sneddon was born on July 7, 1863, in Wishaw, Scotland. He came to this country at the age of thirteen and for three years attended the public schools of St. Louis, Mo. At the end of that time he entered the employ of the Carbondale Coal & Coke Co., Carterville, Ill. So successful was his work there that he was given charge of the construction of a forty-mile railroad for the company. Convinced that he possessed engineering ability, he returned to St. Louis to serve an apprenticeship as machinist with the Rankin-Fritsch Foundry & Machine Co.

Toward the end of his apprenticeship he was sent to Indian Territory as erecting engineer, and later to the Tennessee Brewery at Memphis to install power-plant and refrigerating equipment built by his company. When the plant was put in operation he was asked to assume charge and for the next four years held the position of plant engineer. In 1889 he became master mechanic for the Pittsburgh Plate Glass Co. at their works in Crystal City, Mo., with charge of installation of all equipment. Four years later the Rankin-Fritsch Foundry & Machine Co. tendered him the position of general manager with stock interest, which he accepted, assuming full responsibility for the design and building of their line of Corliss engines and other equipment for power plants.

In 1899 he became superintendent in charge of manufacturing of the Stirling Boiler Co., Barberton, Ohio. In 1906 the company had enlarged greatly and the boiler department of Aultman-Taylor Co. was absorbed, at which time Mr. Sneddon became general manager, in charge of all manufacturing, of the Stirling Consolidated Boiler Co. In 1907 a further consolidation was made with The Babcock & Wilcox Co., and Mr. Sneddon assumed the position of general superintendent with headquarters at Bayonne, N. J. Later his duties were broadened to include the vice-presidency of the Pittsburgh Seamless Tube Co., a subsidiary of The Babcock & Wilcox Co., with full charge of all manufacturing operations for these associated companies.

The war interrupted this service in 1915, at which time he was technical

adviser to the British Mission in the United States. During the following year he was a member of the staff of J. P. Morgan & Co., export department, when they were buying and furnishing supplies for the Allied forces. When the United States entered the war he returned to his regular duties and continued his war activities in manufacturing shell forgings, boilers and tubes for the U. S. Navy Shipping Board and essential industries.

Mr. Sneddon was a member of the Engineers' Club of New York, the Machinery Club, and the Masonic order. He became a member of the Society in 1891.

ADOLPH SORGE, JR.

Adolph Sorge, Jr., for twenty-five years general western representative of the Harrison Safety Boiler Works, now known as the H.S.B.W. Cochrane Corporation, died suddenly on May 5, 1921, in Chicago, Ill. Mr. Sorge was born in Hoboken, N. J., on September 28, 1857. He attended Hoboken schools and Stevens Institute of Technology from which he was graduated as a mechanical engineer in 1875. For two years after graduation he acted as an instructor in Stevens. He then became connected with the E. W. Bliss Co., Brooklyn, N. Y.

About 1893 he became general manager of the Wood-Mosaic Co., Rochester, N. Y., and was largely responsible for developments that occurred in the manufacture of prepared wood-mosaic and parquetry flooring. While in Rochester he acted as adviser to George Selden in framing the claims of the famous automobile patent subsequently issued to the latter. From Rochester Mr. Sorge went to Chicago, becoming superintendent of the foundries of the Fraser-Chalmers Co., building mining machinery and steam engines. In 1895 he resigned to become general western representative of the Harrison Safety Boiler Works, later incorporated as the H.S.B.W. Cochrane Corporation. He was the first to see the advantages of heat in the chemical treatment of water for the removal of hardness and the present Sorge-Cochrane hot process water softener is a further development of ideas originally patented by him. He was often consulted on steam-plant problems and his skill in rectifying defective steam-plant fitting was widely known throughout the West.

Mr. Sorge became a member of the Society in 1886. He was one of the founders of the Technical Club of Chicago and was a member of the Western Society of Engineers and secretary of the American Foundrymen's Association. He also belonged to the Engineers' Clubs of New York and Chicago.

BENJAMIN E. D. STAFFORD

Benjamin E. D. Stafford, vice-president and general manager of the Flannery Bolt Co., Pittsburgh, Pa., died on November 30, 1921, in Atlantic City, N. J. Mr. Stafford was born on February 25, 1866, in Brooklyn, N. Y. He was educated in the public schools of Hopedale, Mass., and later studied in New York City.

As a boy he entered the office of Thomas D. Stetson, patent attorney in New York, becoming proficient in drafting work and in 1882 returned to Hopedale to learn the machinist and tool-making trade with the Hopedale Machine

Co., now known as the Draper Corporation. Five years later he was given supervision of the entire tool department.

From 1893 to 1895 Mr. Stafford specialized in shop practice for a number of the largest machinery concerns in the country and later became associated with the B. M. Jones Co. of Boston as an expert in muschet steel and Taylor staybolt irons. In 1900 he became connected with the Ewald Iron Co., Louisville, Ky., covering the entire United States, demonstrating and placing its products. Four years later he became associated with the Flannery Bolt Co., Pittsburgh. In the same year he was made general manager of this firm and in 1917 became vice-president. Mr. Stafford retired from active business about two years ago in an effort to regain his health and went to live on his farm at Millville, N. J.

Mr. Stafford was a member of the American Society for Testing Materials, the National Geographic Society, past-president of the Railway Supply Men's Association and a member of the Masonic fraternity. He became a member of our Society in 1895.

HAROLD W. STEDMAN

Harold W. Stedman, superintendent of sub-stations for the Ohio Power Co., Newark, Ohio, was accidentally killed on June 10, 1921, while investigating trouble in a sub-station in Shawnee, Ohio. Mr. Stedman was born on November 28, 1895, in Waterbury, Conn. He was a graduate of the Crosby high school of Waterbury and of Pratt Institute, Brooklyn. He was formerly connected with the Chase Metal Works, Waterville, Conn., as assistant chief electrician. In January of the present year he transferred to Ohio. Mr. Stedman became a junior member of the Society in 1917.

EDWARD A. SUVERKROP

Edward Albert Suverkrop, for fifteen years a member of the editorial staff of the *American Machinist*, died on July 29, 1921 of septicemia. Mr. Suverkrop was born in Washington, D. C., May 6, 1864, and was educated in Crieff, Scotland, Fettes College, Edinburgh, Scotland, and Bonn, Germany. He served an apprenticeship in the Canal Basin Foundry, Glasgow, Scotland, from 1882 to 1887 and as a marine engineer visited practically all parts of the world during the next five years. From 1892 to 1894 he was associated successively with the Cramps Shipyard, Philadelphia, Gloucester Manufacturing Co., Gloucester, N. J., and Geometric Lathe Co., Philadelphia. In 1894 he started a machine shop in Camden, N. J., opening a larger one in Philadelphia in 1897. In April, 1904, he became associated with the *American Machinist*. His writings covered a wide field and were under both his own name and the pen name of E. A. Dixie.

Mr. Suverkrop was much interested in the Johansson block and patented in the United States and Europe machinery for manufacturing an accurate gage block. He resigned from the *American Machinist* in July 1919, to develop this method and had begun to place his product on the market at the time of his death.

Mr. Suverkrop became a member of the Society in 1913. He belonged to the Masonic fraternity.

MATTHEW MARK SWEENEY

Matthew M. Sweeney, secretary and general manager of the Fairmount Foundry & Engineering Co., Woonsocket, R. I., died on September 8, 1921. Mr. Sweeney was born in Woonsocket in January, 1879. He was educated in the schools of that city and attended Brown University where he received his B.S. in mechanical engineering in 1906.

Upon graduation he was employed by the Taft-Pierce Manufacturing Co., Woonsocket, as draftsman and machinist in their manufacturing department. From 1908 to 1910 he was connected first with the Bresnahan Show Machinery Co. and later with the General Electric Co., both in West Lynn, Mass., as tool designer. At the end of that period he became production engineer with the General Fire Extinguisher Co., Providence, R. I., where he was located until 1914 when he became associated with the Woonsocket Machine & Press Co. as chief draftsman, resigning to help in the establishment of the Fairmount Foundry & Engineering Co. He served as secretary and general manager of the plant until the time of his death.

Mr. Sweeney was a member of a number of Woonsocket organizations. He became a member of our Society in 1914.

GEORGE PARKER SYMONDS

George Parker Symonds, chief of the engineering department of the Alberger Condenser Co., New York, N. Y., died on February 2, 1921. Mr. Symonds was born on March 26, 1872, in Ogdensburg, N. Y. He attended Cornell University receiving his M.E. degree in 1889. Upon graduation he was employed as assistant engineer in the engineering department of the World's Columbian Exposition, Chicago, where he remained until 1895 when he returned to Cornell and spent the following year gaining shop experience. He then became connected with the Worthington Pump Co., as assistant in the condenser department. In 1902 when the Alberger Condenser Co. was incorporated, he accepted the position of chief of the engineering department.

Mr. Parker was a member of the Cornell Club. He became a member of the Society in 1908.

WILLIAM H. VAN DERVOORT

William H. Van Dervoort, prominent war-munitions manufacturer and president of the Root and Van Dervoort Engineering Co., East Moline, Ill., died on February 25, 1921. Mr. Van Dervoort was born on February 28, 1869, in Ypsilanti, Mich. He attended Michigan Agricultural College as a student in the mechanical engineering department. Following his graduation in 1889, he entered Cornell University for post-graduate work. He returned to Michigan as an instructor in mechanical engineering and continued in that capacity until 1893 when he accepted a position in the mechanical engineering department of the University of Illinois.

In 1899 Mr. Van Dervoort formed a partnership with O. J. Root, a former classmate, for the manufacture of gasoline engines and locomotive specialties. Two years later they moved from Champaign to East Moline. The firm prospered and in 1903 the Moline Automobile Co. was also organized with Mr. Van Dervoort as its president and general manager.

After the close of the war he was chosen as one of a commission of five sent by the National Industrial Conference Board to Europe to study industrial conditions. This commission spent ten weeks in Europe, returning in May 1919. The trip involved many actual physical hardships and Mr. Van Dervoort returned showing visibly the effects of his experiences and with his health severely impaired.

Mr. Van Dervoort was widely recognized as an authority on mechanical engineering and was the author of a book on Machine Shop Tools and Shop Practice. He was also the author of many articles in scientific and technical publications.

He served two years successively as president of the National Metal Trades Association, an honor unique in the annals of that organization. He also served as president of the Society of Automotive Engineers. He was appointed a member of the Munitions Standards Board soon after we entered the war and was selected by manufacturers to act as a member of the National War Labor Conference Board. Later he was appointed by President Wilson to the War Labor Board.

Mr. Van Dervoort became a member of our Society in 1914. He was also a member of other engineering and scientific associations. He belonged to the Engineers' Club of New York City, the University Club of Chicago and was a Knight Templar in the Masonic fraternity.

HARRY R. WARNOCK

Harry R. Warnock, vice-president and general manager of the mechanical department for the Standard Stoker Co., New York, N. Y., died on January 19, 1921. Mr. Warnock was born on July 16, 1870, in Newcastle, Pa., and was educated in the public schools there.

He began his career at the age of seventeen with the Pittsburgh & Lake Erie Railroad Co. as a brakeman, shortly becoming locomotive fireman and engineer and then engine dispatcher at McKees Rocks Shops where he served for two years. At the end of that time he was transferred to a small terminal at Glassport, Pa., as general foreman from which position he resigned to accept a similar position with the West Side Belt Railroad Co. at Pittsburgh, Pa.

In the fall of 1904 he was made master mechanic of the Monongahela Railroad Co. at Brownsville, Pa. where he served for eight years, resigning to become superintendent of motive power with the Western Maryland Railroad at Hagerstown, Md. In December 1917, he was appointed general superintendent of motive power of the Chicago, Milwaukee & St. Paul Railroad Co. at Chicago, and was located there for three years when he accepted the position of vice-president and general manager of the mechanical department for the Standard Stoker Co. He had been with the firm but two months when he was taken ill.

Mr. Warnock became a member of the Society in 1920. He belonged to the Railway Clubs of Pittsburgh and Chicago and to the Masonic Order.

SAMUEL STORROW WEBBER

Samuel Storrow Webber, life member of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS since 1880 and manager from 1902 to 1905, died on April 27, 1921. Mr. Webber was born in Springfield, Mass., on March 31, 1854.

When sixteen years of age he entered the drafting department of the Rogers Locomotive Works, Paterson, N. J. He was next associated with the Blood Manufacturing Works at Manchester, where he became acquainted with the practical side of the business. As mechanical engineer, Mr. Webber accompanied several expeditions for gold dredging in California and South America. He was connected with the Trenton, N. J., Iron Works for twenty-five years, retiring in 1914.

Mr. Webber became a member of the Society in 1880. He was a member of the Engineers' Club of New York.

ALFONSO R. J. WIEDMANN

Alfonso R. J. Wiedmann of the engineering department of the *N. Y. World*, New York City, died on April 17, 1921. Mr. Wiedmann was born in New York City on October 28, 1897. He attended the public schools of the city, New York Preparatory School and Brooklyn Polytechnic Institute from which he was graduated in 1919 with the degree of M.E. He became connected with the Plant Engineering & Equipment Co. in their technical department and subsequently with the *World*, where he was located at the time of his death. Mr. Wiedmann became a junior member of the Society in 1919.

PRESTON KING YATES

Preston K. Yates, consulting engineer, New York City, died on April 22, 1921, in Albany, N. Y. Mr. Yates was born in Canajoharie, N. Y., in 1856. He attended Rensselaer Polytechnic Institute from which he was graduated in 1880. For four years after graduation he was employed in general railroad construction work. From 1884 to 1887 he was assistant engineer on the Croton Aqueduct and for the next three years was chief engineer of a water and sewer company building systems in the South.

Mr. Yates superintended the construction of the 155th St. Viaduct, the new McCombs Dam bridge, the Harlem Ship Canal bridge and the New York Central bridge across Harlem, N. Y. For two years he was associated with the Washburn-Washburn Contracting Co. on the metropolitan water supply and sewerage system of Massachusetts. From 1898 to 1900 he was engaged in the contracting business when he became manager of the L. P. & J. A. Smith Co., Cleveland, Ohio, where he was located until 1904, his duties having to do with the building of docks, breakwaters, piers, power stations, etc. At the close of that period he became first assistant engineer of the Lake Shore Railroad for four years when he again entered into the contracting business, designing and erecting plants for the Tompkins Cove Co., the Texas Trap Rock Co., the New Jersey Trap Rock Co., the Thurber Earthen Products Co., etc. At the time of his death Mr. Yates had just completed a plant for the Highway Commission of the State of Georgia.

Mr. Yates was a member of the American Society of Civil Engineers, the Ohio Society, the Alumni Association of Rensselaer and the Republican Club of Brooklyn. He became a member of our Society in 1914.

SAFETY CODES FOR ELEVATORS

In January 1921 the Council approved a Code of Safety Standards for the Construction, Operation and Maintenance of Elevators, Dumbwaiters and Escalators prepared by The American Society of Mechanical Engineers with the assistance of representatives of the U.S. Bureau of Standards, Elevator Manufacturers Association of the U.S., Elevator Manufacturers Association of N. Y., Casualty and Fire Insurance Companies, American Institute of Architects, and various related engineering societies, elevator specialty manufacturers and independent engineers. This is available in pamphlet form.

POWER TEST CODES

During the year 1921 the Power Test Codes Individual Committees on General Instructions and Reciprocating Steam Engines completed the revisions of these A.S.M.E. test codes. They have passed through the many stages of development required by the Society and are now available in pamphlet form.

A General Committee of 125 specialists have been engaged in revising the A.S.M.E. Power Test Codes of 1915 for the past three years. The two codes now complete constitute the first instalment of the nineteen codes which are in course of preparation by a corresponding number of Individual Committees.

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